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U. S. S. *OMAHA.*

DESCRIPTION OF MACHINERY INSTALLATION AND OFFICIAL TRIALS.

BY LIEUT. COMDR. J. G. B. GROMER, U. S. N., MEMBER.
BOARD OF INSPECTION AND SURVEY, PACIFIC COAST.

The U. S. S. *Omaha,* originally known as Scout Cruiser No. 4, was built by the Todd Drydock and Construction Corporation, Tacoma, Washington, under the authorization of the Acts of Congress dated August 29, 1916, and March 4, 1917, authorizing the construction of seven (7) scout cruisers under "Increase of the Navy."

The original contract for the construction of the *Omaha* was made on December 26, 1916, with the Seattle Construc-

tion and Drydock Company, Seattle, Washington, for a fixed price. Due to the fact that war had been declared on April 6, 1917, and shipbuilding costs had mounted excessively, in order to speed up construction on these vessels, a supplementary contract, dated August 27, 1917, changed the contract price of the *Omaha* from a fixed price of two million five hundred thousand dollars ($2,500,000) to cost plus ten (10) per cent for profit, and two additional scout cruisers, Nos. 5 and 6, the *Milwaukee* and *Cincinnati,* respectively, were ordered from the same company.

In the meantime the keel of the *Omaha* had been laid at the contractor's yard in Seattle, Washington. The Seattle Construction and Drydock Company's plant was bought by the Todd Shipyards Corporation and later the plant was commandeered by the Emergency Fleet Corporation and turned over to Skinner & Eddy for operation. Work on the three scout cruisers was entirely suspended, preference being given to the building of Merchant vessels.

On November 19, 1918, the Secretary of the Navy declared the above contracts forfeited on the part of the Seattle Construction and Drydock Company on account of its failure to proceed diligently with the work of construction.

The contract for the completion of the *Omaha, Milwaukee* and *Cincinnati,* was awarded to the Todd Drydock and Construction Corporation on February 21, 1919, on the basis of cost, including cost of all material and work done prior to forfeiture by the Seattle Construction and Drydock Company, of six million five hundred thousand dollars ($6,500,000), plus a fixed profit, and with an additional bonus of a certain percentage of the cost less than six million five hundred thousand dollars ($6,500,000). The keels of the three vessels were then taken up and relaid at the plant of the Todd Drydock and Construction Corporation, Tacoma, Washington, and all material that had been assembled in Seattle, Washington, shipped to the Tacoma plant.

Later Congress authorized the increase of the limit of cost to eight million five hundred thousand dollars ($8,500,000).

The keel of the *Omaha* was last laid at Tacoma on December 6, 1918, and the vessel launched December 14, 1920.

In accordance with an amendment to the contract, a builder's trial of four (4) hours, at a speed of about twenty-seven (27) knots was held, in deep water in Puget Sound, on January 30, 1923. The vessel was delivered to the Navy yard, Puget Sound, on February 24, 1923, and commissioned the same day. The above amendment required that full trials should be held during the six months and ten days guarantee period, the trials to be run by a Navy crew, and the contractors take all responsibility for penalties or bonuses for performance.

After commissioning, during the guarantee period, the *Omaha* steamed a total of 9,720.7 miles on a "shakedown" cruise, and the full trials authorized for one of the vessels of this class, by the modified contract, with a Navy crew, were held August 11 to 20, 1923. The standardization runs were made on the Vashon Island course and the full speed and fuel oil consumption runs were made in the Straits of Juan de Fuca, in deep water, to the westward of 123 degrees West Longitude.

The reasons for the delay and excess cost above that contemplated by the original plans and contracts were due to the rise in price of shipbuilding during the war, the fact that precedence was given to Merchant shipbuilding and due to the numerous changes made necessary by the attempt to include in these vessels features considered necessary after actual war experience. The hulls were considerably strengthened, more guns, mines, and airplanes added, and the fuel and water capacity was increased.

The three vessels built under this contract are so nearly identical in hull characteristics, battery, boats, complement, etc., with the U. S. S. *Richmond*, described in the JOURNAL

OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, Vol. XXXV, No. 3 of August, 1923, that only the following hull data will be given:

HULL DATA.

Material	Steel
Draft, designed, forward, feet and inches	13–6
Draft, designed, aft, feet and inches	13–6
Draft, designed, mean, feet and inches	13–6
Displacement on designed draft, tons	7,100
Tons per inch immersion	50.6
Length, over all, feet and inches	555–6
Length on load waterline, feet and inches	550–0
Beam, extreme, feet and inches	55–4
Beam on load waterline, feet and inches	54–9
Beam, molded, feet and inches	54–9
At designed draft, area immersed, midship section, square feet	702.4
At designed draft, area L. W. L. plane, square feet	21,340.0
At designed draft, wetted surface, square feet	34,000.0
Length on straight keel, feet and inches	472–4
Depth, molded, main deck at frame 69, feet and inches	26–0
Ratio beam to length	10.045
Coefficient of fineness, block	.611
Coefficient of fineness, midship section	.947
Coefficient of fineness, load waterline	.708
Capacity of engineroom feed tanks, tons	22.32
Capacity of reserve feed water tanks, tons	335.94
Capacity of ship's fresh water tanks, tons	56.00
Number of frames	137
Frame spacing, feet and inches	4–0

MACHINERY LAYOUT.

The machinery layout of the *Omaha* can be readily seen from Plates I and II. There are three boilers in each fireroom, arranged athwartship. The two forward firerooms are forward of the forward engineroom containing the engines for the outboard shafts, Nos. 1 and 4. In the forward engineroom the main condensers are located inboard.

Just abaft the forward engineroom are located the two after firerooms which are similar to the two forward firerooms. Next comes the after engineroom with the two inboard shafts, Nos. 2 and 3, with the main condensers located outboard.

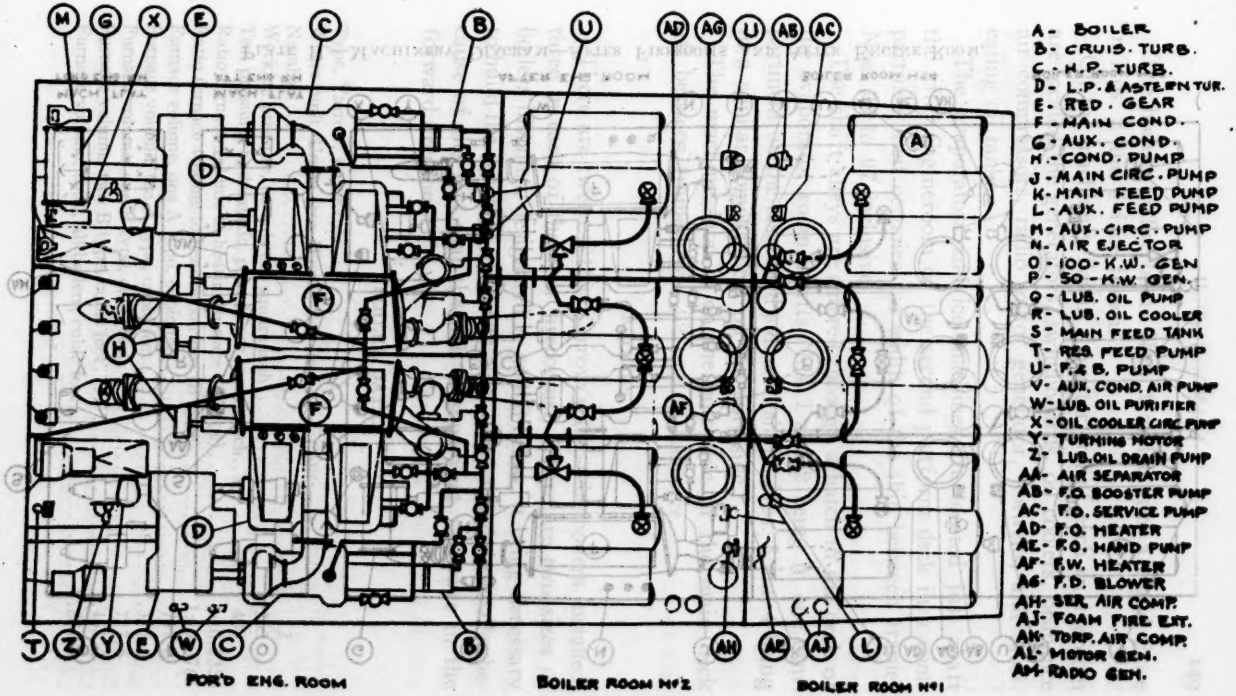

PLATE I.—MACHINERY DIAGRAM—FORWARD FIREROOMS AND FORWARD ENGINE ROOM.

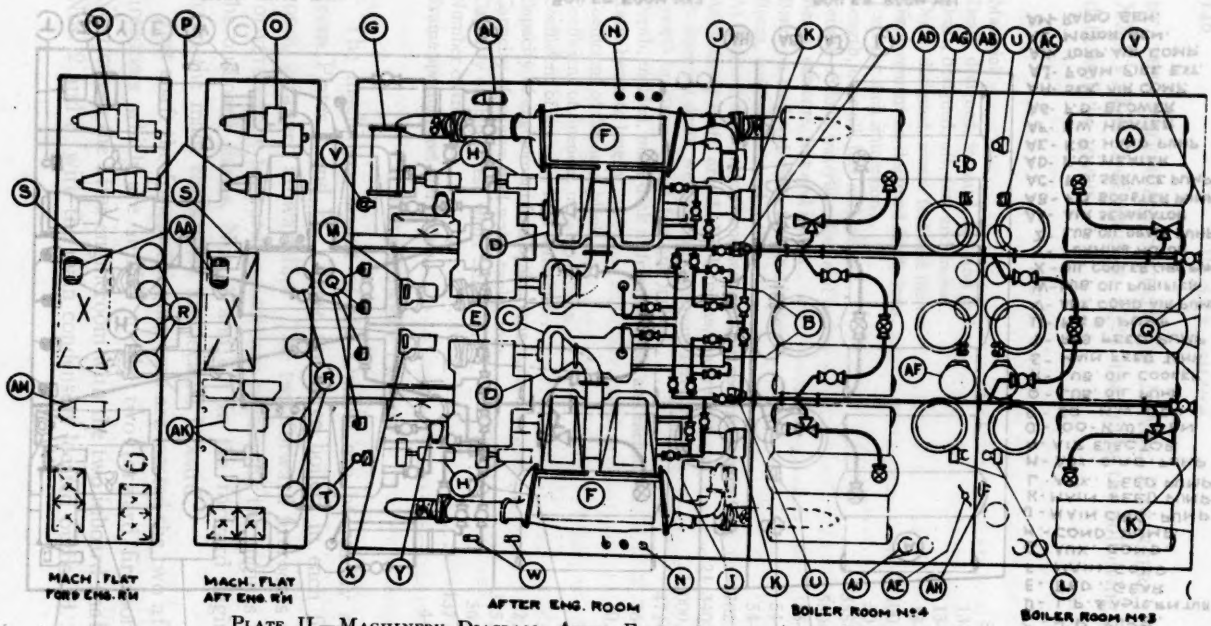

PLATE II.—MACHINERY DIAGRAM—AFTER FIREROOMS AND AFTER ENGINE ROOM.

The boilers in firerooms Nos. 1 and 3 face aft and those in Nos. 2 and 4 face forward.

On the machinery flat in the after end of each engineroom are located the generators, switchboards, oil coolers, ordnance air compressors, radio motor generator, hotwells, and lubricating oil tanks.

The evaporators and ice machines are located just abaft the after engineroom on the second platform deck in a compartment that occupies the width of the ship between frames Nos. 93 to 97 inclusive.

The shaft alleys are entered through a hatch in the deck of the evaporator room, shafts Nos. 1 and 2 being in one compartment on the starboard side and shafts Nos. 3 and 4 being in a similar compartment on the port side.

The machine shop is on the port side of the main deck aft between frames Nos. 92 to 100 inclusive.

BOILERS.

There are twelve (12) Yarrow water-tube boilers installed, three in each of the four firerooms. There are two sizes of boilers installed, designated A and B below, it being necessary to build the furnaces of boilers Nos. 1 and 3 slightly smaller because of the fact that the hull is somewhat narrower in the forward fireroom.

BOILER DATA.

Type, Yarrow, water tube.	
Where built, Union Iron Works, San Francisco, California.	
Number	12
Working pressure, pounds per square inch	265
Test pressure, hydrostatic, pounds per square inch	400
Ratio furnace volume to heating surface (A type)	1:10.2359
Ratio furnace volume to heating surface (B type)	1:10.3765
Furnace volume, one A type, cubic feet	737.6
Furnace volume, ten A type, cubic feet	7,376.0
Furnace volume, one B type, cubic feet	727.6
Furnace volume, two B type, cubic feet	1,455.2
Furnace volume, total, all boilers, cubic feet	8,831.2

Type burners, Lodi B. & W.	
Number of steam drums, each boiler	1
Number of water drums, each boiler	2
Inside diameter, steam drums, feet and inches	4–2
Inside diameter, water drums, feet and inches	2–1
Heating surface, one boiler, square feet	7,550.0
Heating surface, all boilers, square feet	90,600.0
Diameter of tubes, smaller, inches	1
Diameter of tubes, larger, inches	1½
Thickness of tubes, smaller, mils	95
Thickness of tubes, larger, mils	120
Length of tubes, maximum, feet and inches	8–7¾
Length of tubes, minimum, feet and inches	7–6¾
Tube material, cold drawn seamless steel.	

There are four smokepipes, each with an area of 67.2 square feet. The height of Nos. 1 and 2 above the furnace floor is 77 feet 5⅛ inches, No. 3, 71 feet 6⅞ inches, and No. 4, 71 feet 9⅞ inches. The ratio of the heating surface to the area of the connected smokepipe is 337.05.

FORCED DRAFT SYSTEM.

There are twelve (12) B. F. Sturtevant Co's. vertical, turbo-blowers installed, three in each fireroom. Suction is taken through separate vertical armored trunks opening into the space between the drying rooms (uptake compartments) on the main deck. Plates I and II show their location with reference to the boilers in the different firerooms.

BLOWER DATA.

Maker, B. F. Sturtevant Company, Boston, Mass.	
Size and type	VD Type 7
Diameter, bucket wheel, feet and inches	2–6
Width buckets, inches	2¾
Number buckets	80
Number nozzles and reverse buckets	8
Diameter nozzle throat, inch	$\frac{11}{32}$
Area nozzle throat, square inch	.09281
Nozzle mouth, inch, square	½
Number of reversals in each reverse bucket	4
Clearance between bucket and nozzles, inch	$\frac{1}{32}$
Diameter steam inlet to governor, inches	2

Diameter exhaust outlet, inches	4
Diameter relief valve, inches	2½
Type of governor, spring loaded fly ball, speed regulation.	
Bearing lubrication, self-contained forced feed.	
Lubricating oil pump	Gear
Type of fan	Turbo-vane
Diameter fan wheel, inches	48
Designed—	
Steam pressure, pounds per square inch	175 to 300
Exhaust, pounds absolute	0 to 30
Revolutions per minute, normal	1,350
Revolutions per minute, maximum	1,700
Horsepower (normal R. P. M.)	100
Steam consumption, pounds per B. H. P. hour	62½
Capacity fan, cubic feet per minute	50,000
Static head, inches, water	7

FEED WATER HEATERS.

Four (4) Schutte-Koerting, No. 300 Navy Type, vertical straight tube feed water heaters are installed, one in each fireroom. Their characteristics are as follows:

Manufacturer, Schutte-Koerting, Philadelphia, Pa.	
Number	4
Tubes, number	650
Tubes, thickness, mils	49
Tubes, outside diameter, inches	⅝
Tubes, length, feet and inches	7–4¼
Heating surface, square feet, each	744
Heating surface, square feet, total	2,976

FUEL OIL HEATERS.

Twelve (12) Griscom-Russell Co. No. 8 Reilly Navy Type, fuel oil heaters are installed, three (3) in each fireroom. The following data shows their size and heating capacity:

Manufacturer, Griscom-Russell Co., New York, N. Y.	
Number, total	12
Number, each fireroom	3
Number coils	8
Coils, thickness, mils	65
Coils, outside diameter, inches	1
Coils, length, feet and inches	3–9¼
Heating surface, each, square feet	49.2
Heating surface, total, square feet	1,590.4

MAIN PROPELLING MACHINERY.

Each of the four (4) main propelling units of the *Omaha* consist of a high pressure and a low pressure turbine, placed side by side and connected to the line shafting and propeller by a Westinghouse-MacAlpine type reduction gear placed directly astern of the turbines. Plates III shows a cross section of the cruising and H.P. turbines and Plate IV shows a cross section of the L.P. and astern turbines.

The steam passes first through the high pressure unit and thence through a cross-over pipe to the low pressure turbine. The low pressure turbine is of the double flow type, the steam entering the inlet belt at the center and expanding toward both ends, then through the exhaust trunks to the condenser.

The turbine glands are fitted with both steam and water seals, the steam seal being used for all speeds up to half speed and the water seal for all speeds above half speed.

For cruising speeds up to 17.5 knots, a cruising turbine, together with a positive clutch and cruising turbine reduction gear, is installed on each shaft. When using the cruising combination, steam first passes through the cruising turbine, then through a cruising belt of blading in the high pressure turbine and then through the high pressure and low pressure turbines in the usual way.

For speeds above 17.5 knots, the cruising turbine is unclutched and the cruising turbine belt in the high pressure turbine is by-passed.

The main turbines are guarded against damage due to unforeseen accident by fly ball governors of standard Westinghouse design, which control governor valves in the main steam lines, through the medium of oil relays.

The Westinghouse-MacAlpine type reduction gears have pinions which are flexibly supported so that an even distribution of the load on the gear teeth is obtained.

Each main turbine unit is served by a single pass surface condenser supplied with cooling water by means of scoop in-

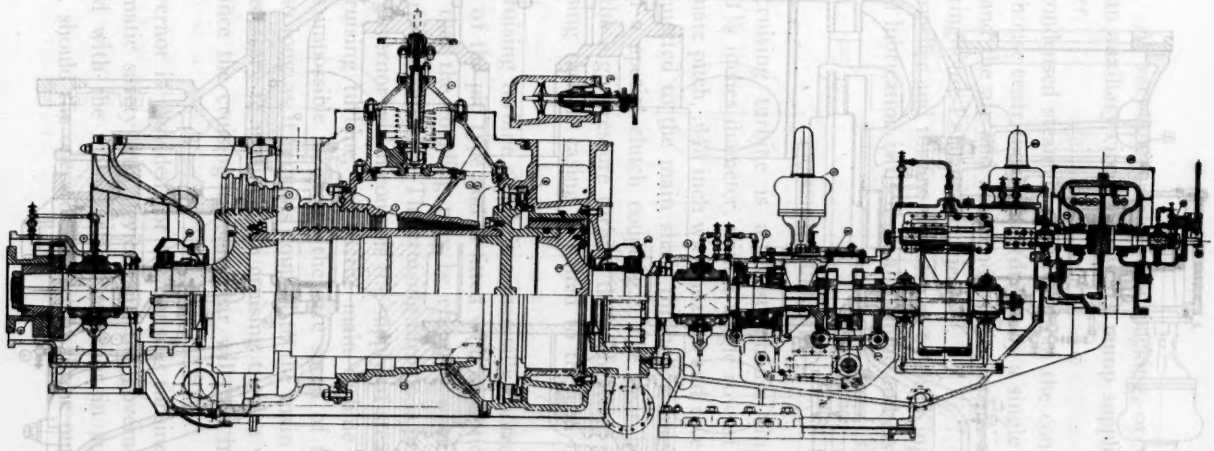

PLATE III.—CROSS-SECTION OF CRUISING AND H.P. TURBINES.

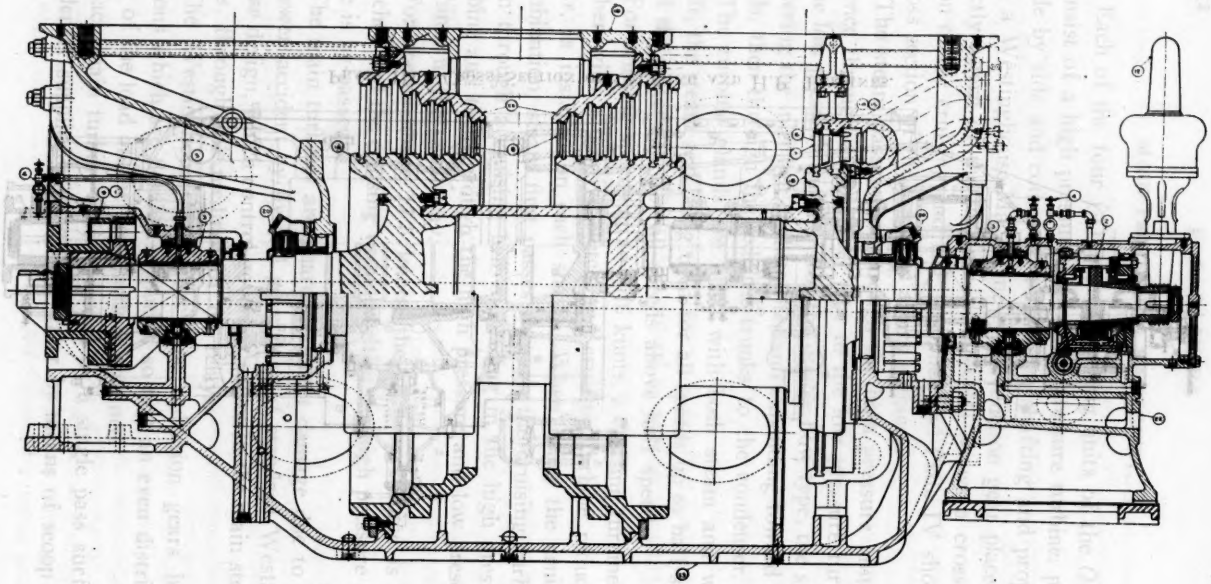

PLATE IV.—CROSS-SECTION OF L.P. AND ASTERN TURBINES.

jections. At very low speeds, when backing, or when standing by, an auxiliary circulating water pump supplies the cooling water.

The condensed steam is removed from the condenser by a Westinghouse centrifugal pump driven by a single stage steam turbine and the air and non-condensable vapors are removed by Westinghouse-LeBlanc air ejectors.

All propeller and turbine thrusts are of the Kingsbury type.

Forced lubrication is used on all turbine and reduction gear bearings.

CRUISING TURBINE.

The cruising turbine is a Westinghouse single impulse wheel, 21⅛ inches diameter, with a single row of 128 blades, approximate pitch, .645 inch with nozzles and reversing chambers, connected to the main shaft through a single reduction gear and a special clutch coupling. The gear housing is a part of the high pressure turbine thrust bearing pedestal and the cruising turbine casing is bolted to the end of this gear housing.

The cruising turbine clutch is a manually controlled sliding coupling of the pin type and is fitted with an optical synchronizer so that the coupling may be engaged when the speeds of the turbines coincide. The cruising turbine is provided with a safety governor which controls the supply of steam, not only to the cruising turbine, but to the main turbine as well, so that it is impossible to exceed the safe speed of the cruising turbine so long as it remains coupled to the main engine. It is not necessary to unclutch the cruising turbine when going astern since the cruising turbine gear will run in either direction.

A governor is provided for the cruising turbine as well as an automatic safety stop device, and the governor is interconnected with the main turbine governor in such a manner that, should the operator neglect to throw out the clutch

when cruising turbine speed limit is reached, the cruising turbine governor would shut down both turbines and prevent dangerous overspeeding of the smaller unit.

TURBINE DETAILS.

Expansions	Diameter of cylinder, feet and inches		Length of cylinder, (this diameter)		Rows of blading		Pitch of blades	
	Turbine		Turbine		Turbine		Turbine	
	H. P.	L. P.	H. P.	L. P.	H. P.	L. P.	H. P.	L. P.
			Inches	*Inches*				
1	4–1⅛	4–4	5 13/32	2⅛	2	1	...	.435
2	4–2½	4–6	5 9/32	2⅜	2	1	...	.683
3	2–7½	4–9	1⅞	5⅛	2	2	.237	.737
4	2–8	5–0	3	2¾	4	1	.237	.737
5	2–9	5–3	4 7/16	6½	5	2	.237	.977 .752
6	2–10	...	4	...	4	...	.237	...
7	2–11	...	5	...	4	...	.333	...
8	3–0	...	3½	...	2	...	.435	...
9	3–2	...	6⅞	...	3	...	.683	...
10	3–4	...	9⅛	...	4	...	.737	...
11	4–0	...	5⅛	...	2	...	.683	...
12	4–2	...	2⅜	...	1	...	.683	...
13	4–5	...	6⅛	...	2	...	.977	...

NOTE:—Stages Nos. 1 and 2 of the H.P. turbine are impulse blading. Astern turbine, diameter of cylinder 4 feet 2¾ inches, length 6-15/16 inches, two rows of blading 6⅝ inches long. The cruising turbine, Westinghouse, single impulse wheel, 21⅛ inches diameter, single row of 128 blades, approximate pitch .645 inch, with nozzles and reversing chambers.

TURBINE OPERATION.

Going Ahead—Steam from the ahead throttle valve is admitted to the 12 inch opening in the top and to one 6 inch and one 3 inch opening in the bottom of the high pressure turbine, Plate III. These openings lead into steam chambers in the head of the turbine and steam passes from them through control valves in the cylinder head, into the various nozzle groups. The steam expands in the nozzle and passes through the impulse blades and thence through the reaction blading. Having passed out of the exhaust pipe of the H.P. turbine it goes through a suitable connection pipe into the L.P. turbine, Plate IV, entering at the center and flowing both ways through the reaction blading.

The number of hand nozzle control valves to be opened depends on the load as does also the opening of the by-pass valve.

The following table gives the hand valves, each of which has a number, to be open for the standard speeds shown:

Standard Speed Knots	Hand Nozzle Control Valves Open-Numbers	By-pass Valves
Maximum	2-3-4-5-6-7-	Open
35	3-4-5-6-7	Open
30	5-6-7	Open
25	5	Closed
20	7	Closed
15	1	Closed
CRUISING TURBINE.		
17-1/2	1	Closed
15	1	Closed
12	2	Closed

The nozzle groups controlled by the hand nozzles have the following percentages of the total nozzle area, which is a direct measure of the amount of steam which will flow and therefore approximately, of the load.

Nozzle No.	1	2	3	4	5	6	7
Per cent of total area (approx.)	6	16	18	18	19	13	9

There is one small nozzle in the bottom between the groups controlled by valves 1 and 7 which is open direct to the 6 inch inlet so that some steam will blow through the bottom of the cylinder when valves 1 and 7 are closed. This nozzle is less than 1 per cent of the total.

By the proper use of these hand valves for the different standard speeds, adjusting the nozzle area to the load to be carried, the maximum pressure is maintained at the nozzle inlets, and thus better efficiency is obtained than if the nozzle area were much too large for the load and the steam were throttled to a considerable amount at the inlet. In case pressure or vacuum falls off it may be necessary to open additional nozzles, choosing the valve which will give only about the additional load needed. Thus, if No. 5 were open for 25 knots standard speed and full speed were called for with the steam pressure somewhat low, number one valve would be opened, which would add about 20 per cent to the available nozzle area.

Speed is of course, regulated from the throttle valve only, in the usual manner, the use of the hand valves being to get the best economy and to prevent overloading of the boilers at the different standard speeds. For maneuvering in harbor, etc., it is advisable to have nozzles opened which will give about twice the area required by the standard speed set, so that quick changes of speed can be made.

Going Astern—In the forward end of the low pressure turbine, Plate IV, is located the astern turbine which is a single two row impulse wheel supplied with steam from nozzles in four chambers. Each of these chambers is supplied through a 6-inch inlet in the head of the cylinder, these four inlets being connected to a single astern throttle valve. As this astern wheel is in the exhaust chamber of the main turbine it revolves in a vacuum when running ahead, as does the ahead portion

when running astern. There being only a single impulse wheel in the astern turbine, steam may be admitted as rapidly as desired after shutting off the steam to the ahead turbine.

BALANCE OR DUMMY PISTON.

When under steam, the H.P. turbine, due to its construction is subjected to a steam thrust tending to move the rotor in a fore and aft direction. This steam thrust is partly balanced by a balance piston or dummy, shown on Plate III, fitted at the ahead steam inlet end of the turbine. The back of this dummy is connected through a pipe to the exhaust end of the turbine. The steam thrust on the turbine rotor is by this means either completely or partially balanced depending on the diameter or area of the dummy piston. To balance this steam thrust completely would necessitate the use of a larger dummy piston than is fitted to these machines. The smaller the dummy can be made the more rugged and reliable is the turbine. Hence, it is this Company's practice only partially to balance the steam thrust on the turbine rotor with a small dummy and to support the remaining thrust on a pivot thrust bearing of the Kingsbury type. The dummy piston is formed by a forged steel ring shrunk and keyed on to the turbine spindle. On this ring, which is a casting fitted to the cylinder, are fitted aluminum strips. These strips have a thin edge which comes close up to the ridges turned on the turbine spindle dummy ring. As the clearance between them is very small there is little steam leakage from the high pressure end to the low pressure end of the dummy which is connected to the exhaust of the turbine by an equilibrium pipe. For the H.P. turbine this clearance is .010 inch.

BLADING.

The Westinghouse method of securing impulse blades is shown in Plate V, the purpose of the wedges being to insure

the blades being held tightly in the slot, and to preclude any vibration being conveyed to the reduced section that is below the surface of the blade carrying element.

The method of attaching reaction blading is to provide a supplementary groove in the floor of the main groove which receives an upset foot on the blade. This upsetting operation increases the thickness of the whole lower portion of the blade, adding to its area about 40 per cent, tapering off to the standard section some distance above the surface of the blade carrying element. Dovetailed packing pieces come between the blades, and the forged feet of the blades coming beneath the

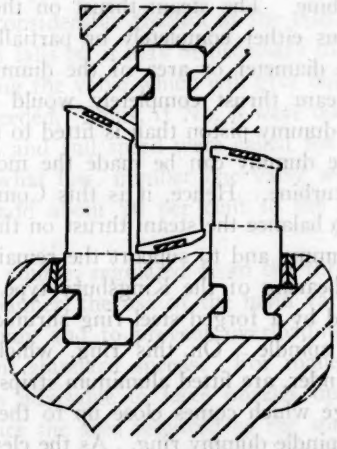

PLATE V.—METHOD OF SECURING IMPULSE BLADING.

packing pieces provides an interlocking system with which no caulking is necessary, making it possible to replace blades an indefinite number of times without mutilation of the blade carrying member.

Lashing wires in sufficient number are provided for bracing the upper ends of the blades, and the number of these lashings depends on the dimensions of the blades.

The astern nozzle chambers, are independent castings. The root section of the stationary blades, between the first and second rows of rotating impulse blades, is attached to the nozzle chamber. The method of attachment to the main cylinder casting allows for the free expansion and contraction of the nozzle chamber independently of the main cylinder casting.

The ahead nozzle chambers are integral with the cylinder casting, but the methods of attachment of nozzles and blades are the same.

GLANDS.

The glands for preventing leakage where the rotor emerges from the casing on both the high and low pressure turbines are of the combined steam and water sealed type. The water gland is designed to effectively seal at all speeds above half speed. Below half speed the steam seal is used.

Steam is supplied to this gland at the 1-1/2-inch pipe tap for steam connection and water at either one of the 1 inch connections for water. The steam pressure is regulated by a reducing valve in the pipe line to the glands while the water is supplied from a small connection from the feed line. This connection has a small orifice at some convenient point and an overflow to the feed tank through a relief valve set at about 7 pounds. The method of controlling the supply of water and steam to the glands during changes in speeds is described under gland control valve.

The water gland runners are designed similar to the runners of a centrifugal pump. They are shrunk on and doweled to the turbine shaft. At one-third speed and above the runners would, if used as a pump and supplied with water at the inner radius, discharge the water to a level greater than that corresponding to the water pressure. Due to this pumping capacity of the gland runner the water which is supplied to the periphery of the runner is unable to either enter the turbine or escape on the outside of the runner to the atmos-

phere. Thus there is maintained at the outer edge of the runner a solid annulus of water producing an hermetic seal entirely precluding leakage. Any water that may leak out of the gland on the outside is carried away by a drain.

The L.P. steam seal consists of two sets of aluminum packing rings. These rings are machined with projections which engage corresponding shoulders in the gland casings. Flat springs are fitted at the back of these rings to hold them out against the shoulders in the casings. The points of the rings are made a little long so that the projection on the ring does not at first engage the shoulder in the gland casing. A little running causes the points to wear slightly until the contact at the shoulder is made, after which no further wear can take place and the rings form a close and effective seal around the turbine shaft.

The H.P. pressure steam seal is slightly different in design from the L.P. seal, as it has to pack against a back pressure greater than the atmosphere. As the back pressure in the turbine is greater than the pressure due to the head of water on the water wheel chamber, steam will leak out past the first rows of packing rings. To prevent the escape of this steam to the engineroom a leak off chamber is provided between the two sets. This chamber is piped to the condenser.

GLAND CONTROL VALVE.

As the turbine glands are of the combination type and since the water seal is not effective below half speed, it is necessary to change over from steam to water or vice versa, whenever the half speed point is passed.

In order to relieve the operator of the necessity for making this change by hand, an automatic gland control valve is installed.

The control cylinder is connected to the governor oil-relay by a pipe from the bottom of the cylinder and a pipe from the top leads to the drain tank. The steam and water

valves are placed side by side and are operated by a lever and piston rod. The steam valve is opened by upward motion and the water valve by downward motion so that it is impossible for both to be either open or shut simultaneously.

When the turbines are at rest, or running at less than half speed, the governor holds the relay in such a position that no oil pressure is communicated to the under face of the control piston. Hence, the piston remains at the bottom of its travel and the steam valve is open and the water valve shut.

When the turbine exceeds half speed the movement of the governor relay plunger opens the connection to the operating cylinder and admits oil under the piston. This of course causes it to move upward, opening the water valve and closing the steam valve.

GOVERNOR AND GOVERNOR RELAY.

The governor is of the ordinary simple spring and fly-ball type, operated by means of a worm and gear from the main turbine spindle. Two springs are used, a heavy one and a light one. For the first half of the travel the weights are opposed only by the light spring, the heavy spring resting against the supporting piece. When the turbine reaches about one-third speed the centrifugal force on the weight is sufficient to balance the pressure on the light spring and the weights move outward, lifting the yoke until it comes in contact with the spring seat ring on which rests the heavy spring.

This motion moves the relay plunger downward, uncovering the port communicating with the connection to the gland control valve or change-over valve previously described, causing it to shift over from steam to water.

As the turbine speeds up the governor remains in this middle position until the centrifugal force is sufficient to overcome the combined pressures of the light and heavy springs, which will occur at about 5 per cent to 7 per cent above full speed. The governor weight will again move outward and

again lower the plunger until the upper port, communicating with governor valve opens, admitting the oil to the lower end of the relay on the governor valve and causing it to close, as described under governor valve.

Oil is admitted to the governor through a 3/8-inch pipe tap and enters the chamber around the governor spindle bushing and passes through holes drilled in same into grooves along the governor spindle. Oil flows downward out through the bottom bushing and up through the upper bushing, lubricating the thrust plate through suitable small grooves in the bushing itself. It also flows through suitable holes and lubricates the governor clutch bushing and also the governor clutch. The oil which escapes out past the thrust washer passes down through the cored passage in the governor case into the open chamber, where the amount flowing may be readily observed.

The governor bushing is stationary and the governor spindle revolves inside of it while the governor clutch bushing revolves outside of it. Thus all of the parts which will move in response to a change in the speed are already moving with respect to each other in a rotational direction, so that there is minimum friction to oppose a change in speed.

The relay plunger is held in its position through a yoke and rod by a spring, so that it may be operated by hand by compressing the spring, and allowing the plunger to move. Thus it can be frequently ascertained that the plunger is free to move and has not become clogged due to foreign matter getting into the close clearance between the plunger and the plunger bushing.

MAIN GEARS.

These gears, shown in Plate VI, are of the type known as the "Westinghouse reduction gears with flexible frames." Each main gear wheel is driven by two pinions, the gear wheels being mounted in bearings rigidly supported by the gear housing while the pinions are carried in bearings fitted with a flexible frame. The object of the flexible frame is, primarily,

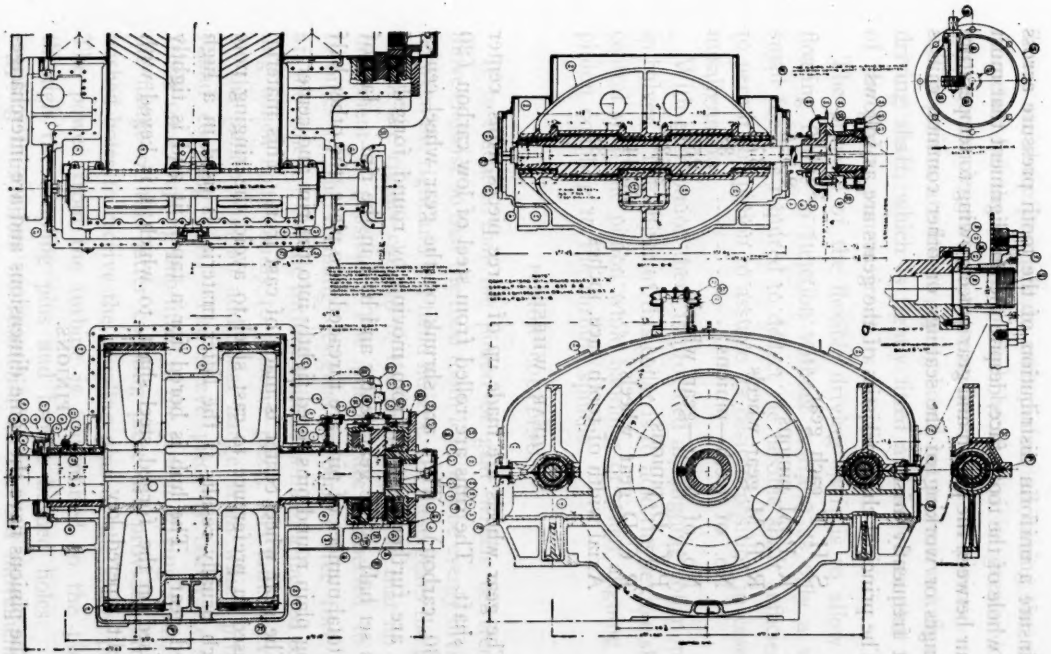

PLATE VI.—MAIN GEARS—WESTINGHOUSE—MACALPINE TYPE.

to insure a uniform distribution of the tooth pressure across the whole of the tooth face, despite the misalignment that must occur between the pinion and gear wheels owing to temperature changes or working of the seatings or other common causes that frequently arise.

The principal characteristics of the gears are as follows:

S. H. P. each gear	22,500
R. P. M. pinions	2,525
R. P. M. gear wheels	370
No. of teeth—pinions	38
No. of teeth—gear wheels	259
P. C. D. pinions	12,537
P. C. D. gear wheels	35,448
Axial width of tooth face, inches	38

GEAR WHEELS.

The gear wheels are made up of three pieces, rims, center and shaft. The rims are rolled from steel of low carbon (20 to 30, carbon). They are shrunk on the gear wheel center and are further secured from moving by round forged steel pins set half in the gear center and half in the rim. Half of the total number of pins are threaded into place, the other half being plain round pins fitted tightly into the rim and center.

The gear wheel center is made of cast iron. This material is used in preference to cast steel to avoid the ringing note which usually results if the latter material is used in a high speed gear. The hub is bored for a taper fit and is tightly pressed on the forged steel shaft, to which it is keyed with a split tapered key.

PINIONS.

The pinions have the same dimensions and are interchangeable. They are forged from carbon steel having a carbon content of between .55 and .65 to insure satisfactory hardness.

There are three journals on each pinion, all having the same diameter, but the center journal, because of its greater load, has a greater length than those at the ends of the pinion.

The pinions are hollow, being bored out to receive a flexible driving shaft, which is tightly fitted and keyed to that end of the pinion remote from its coupling.

The purpose of the flexible driving shaft is to allow the floating frame to function without restraint as only a very small force is required to deflect the flexible shaft sufficiently to permit the pinion to assume its desired position in case of misalignment.

With each pinion there is provided an aligning collar made in halves. The object of this collar is to hold the flexible shaft central with the pinion while handling or when aligning the pinion coupling to the turbine spindle.

PINION COUPLING.

It is essential that the pinion or gear wheels be free to move endwise so that the total tooth load will be evenly divided between the right and left hand helices. In these gears the gear wheels are rigidly coupled to the thrust shaft so that no end movement is possible. The pinions are connected to the turbines by the coupling designed to permit free fore and aft movement. On the turbine spindle is fitted a flanged coupling which bolts solidly to a sleeve piece, this latter having, on the pinion end, an additional flange from which project round coupling pins. The sleeve piece is bored to slide freely on a coupling hub fitted to the pinion driving shaft. In the flange of this latter coupling piece are provided bronze bushed holes to receive the coupling pins of the sleeve piece. The power from the turbine is transmitted to the pinion through the coupling pins and bronze bushed holes. The coupling thus offers no restriction to the end movement of the pinion as the coupling pins are sufficiently long to allow for a considerable end movement and still maintain full con-

tact in the bronze bushed holes through which they drive. To avoid any restriction to end freedom the pinion and gear bearings are fitted with ¼-inch total clearance at each end of each bearing. The pinion and gears can thus move ¼-inch forward or aft of their central position before fouling.

TURBINE COUPLING.

The coupling used for driving a helical gear must permit the pinion of the gear to move in an axial direction so that the load on the teeth may be equally distributed over the right and left hand helices. On the present installation this is accomplished by a pin type coupling, Plate VII. Coupling sleeves (4) and (13) are pressed and keyed on to the turbine spindle and pinion driving shaft respectively. Bolted to the coupling sleeve of the pinion and extending over and centering on the coupling sleeves of the turbine spindle is a coupling head (9). On the turbine end of this head are mounted several pins (10) of suitable size, which engage corresponding bronze bushed holes in the coupling head on the turbine spindle. Clearance is provided at each end of these pins and also in the bronze bushed holes so that the pinion of the gear is perfectly free to slide axially without imposing stress on either turbine or gear. The bronze bushings are lubricated by oil which spills out of the bearing and falls into a centrifugal lubricator formed in the coupling head. Holes are drilled from this lubricator to each bronze bushing. In similar manner, the sliding fit between coupling head and the coupling sleeve at the turbine end is lubricated.

MAIN CONDENSERS.

The main condensers, four in number, are of the bent tube type, with steel shells and composition tube sheets and sea water chests. Both ends of the tubes are expanded in the tube sheet.

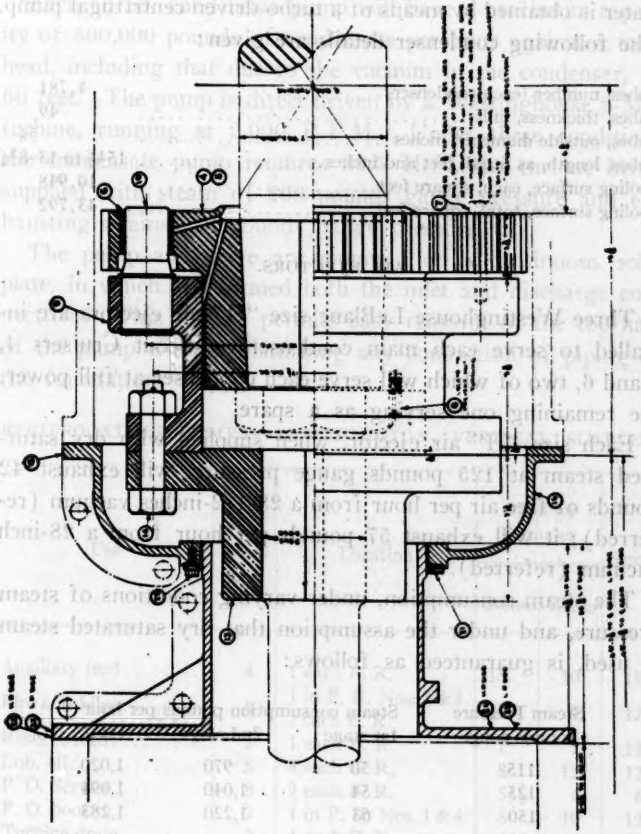

PLATE VII.—TURBINE COUPLING.

Cooling water is supplied through scoops in the ship's bottom and makes a single pass through the tubes. When warming up, running at low speeds or maneuvering, circulating water is obtained by means of a turbo-driven centrifugal pump. The following condenser details are given:

Tubes, number (each condenser)	4,781
Tubes, thickness, mils	49
Tubes, outside diameter, inches	⅝
Tubes, length, as fitted, feet and inches	15½ to 13–8¾
Cooling surface, each, square feet	10,948
Cooling surface, total, square feet	43,792

AIR EJECTORS.

Three Westinghouse LeBlanc size "H" air ejectors are installed to serve each main condenser on Scout Cruisers 4, 5 and 6, two of which will serve each condenser at full power, the remaining one serving as a spare.

Each size "H" air ejector, when supplied with dry saturated steam at 125 pounds gauge pressure, will exhaust 42 pounds of free air per hour from a 28-1/2-inches vacuum (referred); it will exhaust 57 pounds per hour from a 28-inch vacuum (referred).

The steam consumption, under varying conditions of steam pressure, and under the assumption that dry saturated steam is used, is guaranteed as follows:

Steam Pressure pounds gauge	Steam consumption pounds per hour 1st stage	2nd stage	total
115	50	970	1,020
125	54	1,040	1,094
150	63	1,220	1,283

CONDENSATE PUMPS.

Seven Westinghouse, size 33-SN, condensate pumps are installed, three being located in the forward engineroom and four in the after engineroom. In the forward engineroom,

one pump will serve as a spare, while two pumps will serve as spares in the after engineroom. This difference was made necessary owing to the layout of the plant.

Each 33-SN condensate pump will have a maximum capacity of 500,000 pounds of condensate per hour against a total head, including that due to the vacuum in the condenser, of 60 feet. The pump is direct driven by a Westinghouse 2-MD turbine, running at 2,000 R.P.M. Under these conditions the condensate pump requires 36.5 H.P., the turbine being supplied with steam of 200 pounds gauge pressure and exhausting against 10 pounds gauge pressure.

The pump and drive are mounted on a continuous soleplate, in which are formed both the inlet and discharge connections. This feature permits easy removal of the top half of the pump without disturbing any connection of piping or the foundation.

RECIPROCATING PUMPS—WORTHINGTON, VERTICAL SIMPLEX.

Use	Number	Location	Cylinder diameter		Stroke
			Steam	Water	
Auxiliary feed	4	1 each F. R.	15	10	18
Fire and bilge	6	1 in F. R. Nos. 2 & 3 2 each E. R.	10	9	12
Reserve feed	2	1 each E. R.	7	7	12
Lub. oil	8	4 each E. R.	8	12	12
F. O. Service	8	2 each F. R.	5½	4	8
F. O. booster	2	1 in F. R. Nos. 1 & 4	8	10	12
Turbine drain	2	1 each E. R.	4½	8	6
Aux. air	2	1 each E. R.	6½	12	8
Dist. F. W	2	Evap. room	4½	5	6
Dist. air	1	Evap. room	6½	10	8
Evap. brine	1	Evap. room	4½	6	6
Evap. feed	2	Evap. room	4½	6	6

TURBO-CENTRIFUGAL PUMPS—HORIZONTAL.

Use	No.	Make	Location	Capacity
Condensate	7	Westinghouse	3 in for'd E. R. 4 in aft E. R.	500,000 lbs. per hour
Main circulating	4	Worthington	2 each E. R.	10,500 G.P.M.
Main feed	4	Worthington	2 each E. R.	600 G.P.M.
Oil cooler circulating	2	Worthington	1 each E. R.	1,350 G.P.M.
Aux. condenser circulating	2	Worthington	1 each E. R.	1,350 G.P.M.
Dist. circulating	2	Worthington	Evap. room	

MOTOR DRIVEN CENTRIFUGAL PUMPS.

Brine circulating	2	Worthington	Ice machine room	25 G.P.M.
Cooling water circulating	2	Worthington	Ice machine room	25 G.P.M.
Lub. oil drain	2	Todd	1 each E. R.	8 inch

LUBRICATING OIL COOLERS.

Type, Alberger-28P-corrugated straight tube type.	
Number (4 type A and 4 type B)	8
Location—each engineroom—2 each type	4
Cooling surface, type A, each, square feet	280
Cooling surface, type B, each, square feet	360
Cooling surface, total, square feet	2,560
Tubes, number, type A	240
Tubes, number, type B	300
Tube thickness, mils	35
Tubes, outside diameter, inch	$\frac{3}{4}$
Tubes, length, feet and inches	6–2$\frac{5}{8}$

AUXILIARY CONDENSERS.

Type, Todd, bent tube.	
Number	2
Location, one in each engineroom.	
Cooling surface, each, square feet	1,149
Tubes, number	1,109
Tubes, thickness, mils	49
Tubes, outside diameter, inches	$\frac{5}{8}$
Tubes, length, feet and inches	6–5$\frac{13}{16}$

ICE MACHINES.

Two Audiffren-Sigrun, Marine Type, No. 6, ice machines are installed, each having a two (2) ton capacity.

EVAPORATORS.

Four Schutte-Koerting, No. 30, Navy Type, evaporators, each having a heating surface of 114.5 square feet, are installed. Two Schutte-Koerting, bent tube type distillers, each having a cooling surface of 200 square feet are used.

GENERATORS.

There are installed in each engineroom one (1) 50 K.W. and one (1) 100 K.W. General Electric turbo-generators. The voltage used is 125 volts.

MISCELLANEOUS AUXILIARY MACHINERY.

Two, vertical simplex, Westinghouse, air compressors are installed for ship's use, size 11 inches × 11 inches × 12 inches.

Two, DeLaval, motor driven lubricating oil purifiers with a capacity of 100 G.P.H. are installed in each engineroom.

One Gary-Cummings torsion meter is installed on each shaft.

PROPELLERS.

Type	Solid
Material	Comp. Mn-C
Number of blades, each	3
Diameter, feet and inches	11–3
Pitch, feet and inches	11–10
Ratio, diameter to pitch	.950
Area, projected, square feet	60.13
Area, disc, square feet	99.40
Ratio, projected to disc area	.605
Immersion, feet and inches	4–4

OFFICIAL TRIALS.

The trials to be run, with a Navy crew, in accordance with the terms of the modified contract, during the six months and ten days guarantee period were as follows:

(*a*) A progressive trial over a measured mile course in deep water, designated by the Secretary of the Navy, for standardizing the screws, extending from maximum speed down to a speed of nine (9) knots; about twenty-nine (29) runs to be made over the course in order to adequately cover the range of speed desired. This progressive trial is to be made with the vessel weighted as specified hereinafter.

Five consecutive runs must be made at the highest speed attainable and the mean speed and revolutions of these runs carefully ascertained, and three consecutive runs must be made at each of the following speeds, as nearly as possible: Thirty-two and one-half (32½) knots, thirty (30) knots, twenty-seven and one-half (27½) knots, twenty-five (25) knots, twenty-two (22) knots, seventeen (17) knots, twelve (12) knots, and nine (9) knots; the mean speed and revolutions for the several groups of runs to be carefully ascertained by the trial board, in order to determine the curve of speed and revolutions, the vessel to be so weighted that the middle run at the maximum speed will take place, as nearly as may be, at the mean trial displacement specified hereinafter.

(*b*) A full-speed fuel-oil consumption trial of four hours duration in the open sea in deep water, at the highest speed attainable, and the Contractor hereby guarantees that the power developed by the propelling machinery of the vessel upon this trial shall not be less than an average of 90,000 shaft horsepower, to be determined by Gary-Cummings torsion meters connected to the shafting abaft the propelling machinery, thrust bearings, and reduction gears. During this trial the steam pressures in the boilers, piping, and turbines shall not exceed those stated in the machinery specifica-

tions. This full-speed trial is to be made with the vessel weighted as specified hereinafter.

(*c*) An endurance and fuel-oil consumption trial of four (4) hours in the open sea in deep water at an average uniform speed of thirty (30) knots as nearly as possible, the vessel to be weighted as specified hereinafter.

(*d*) A fuel-oil consumption trial of six (6) hours in the open sea in deep water at an average uniform speed of twenty-five (25) knots as nearly as possible, the vessel to be weighted as specified hereinafter.

(*e*) A fuel-oil consumption trial of four (4) hours in the open sea in deep water at an average uniform speed of twenty (20) knots as nearly as possible, the vessel to be weighted as specified hereinafter.

(*f*) An endurance and fuel-oil consumption trial of four (4) hours in the open sea at an average uniform speed of fifteen (15) knots as nearly as possible, with the cruising units connected and in use, the vessel to be weighted as specified hereinafter. This trial shall be made as nearly as possible under service cruising conditions.

(*g*) An endurance and fuel-oil consumption trial of two (2) hours in the open sea at the specified speed of trial (*f*), the cruising units to be disconnected and not in use; the vessel to be weighted as specified hereinafter.

(*h*) A fuel-oil consumption trial of four (4) hours in the open sea at an average uniform speed of ten (10) knots as nearly as possible, and with the cruising units connected and in use; the vessel to be weighted as specified hereinafter.

(*i*) Trials will be conducted to determine the ability of the vessel, when going ahead at full speed, to back satisfactorily at full backing power, this power to be sufficient to give a speed of not less than twenty (20) knots when going astern; to determine the time required to bring the vessel dead in the water and the distance ahead reached when steaming at full

speed; to test the steering gear both when going ahead at full speed and when backing at the speed obtained with the maximum backing power the machinery develops; and to test the anchor gear.

(*j*) An endurance trial of two hours in the open sea at an average uniform speed of seventeen and one-half (17-1/2) knots as nearly as possible, with the cruising units connected and in use, the vessel to be weighted as specified hereinafter. This trial shall be made as nearly as possible under service conditions. This trial will be conducted to determine the ability of the vessel to maintain a speed of seventeen and one-half (17-1/2) knots with the cruising unit connected and in use.

Trials as necessary will be made to test the operation of the clutch between cruising turbines and pinion shafts at speeds up to and including fifteen (15) knots; and trials, as considered necessary, to determine the ability to back promptly when steaming ahead at about fifteen (15) knots with the cruising turbines in use. These trials are particularly intended to determine the efficiency of the methods provided for disconnecting the cruising turbines before backing, or the practicability of running the cruising turbines backward.

On trials (*c*), (*d*), (*e*), (*f*), (*g*), and (*h*) all necessary auxiliaries shall be in operation, including all those usually required under service cruising conditions and especially such dynamos as may be necessary for efficiently lighting the vessel. During the first three hours of trial (*e*) the evaporating and distilling plant will be in operation to produce 800 gallons of fresh water.

On trials (*b*), (*c*), (*d*), (*e*), (*f*), (*g*), and (*h*) there shall be carefully determined and recorded the fuel-oil consumptions and the amounts of water required to be used for make-up feed. During trial (*b*) the steam pressures in the boilers, piping, and turbines shall not exceed those in the machinery specifications.

STANDARDIZATION RUNS.

Run	Knots	Revolutions per minute				Shaft horsepower				Total S.H.P.
		Shaft numbers				Shaft numbers				
		1	2	3	4	1	2	3	4*	
1	9	78.25	78.52	79.35	79.25	194	204	272	175	845
2	9	80.47	83.22	81.45	81.33	200	288	256	200	944
3	9	79.67	81.94	83.08	86.80	173	260	261	260	954
4	12	111.14	110.18	106.03	108.18	588	604	393	530	2,115
5	12	113.22	111.40	110.25	111.85	563	611	440	580	2,194
6	12	112.85	112.04	109.44	111.22	562	646	437	570	2,215
7	17.5	165.76	157.06	156.63	155.99	2,062	1,631	1,386	1,630	6,709
8	17.5	163.25	160.91	159.88	158.20	1,929	1,903	1,688	1,720	7,240
9	17.5	161.07	159.33	161.34	160.20	1,753	1,747	1,750	1,780	7,030
10	22	219.44	214.45	213.41	215.53	4,641	4,825	4,446	4,210	18,122
11	22	214.62	215.51	218.36	217.17	4,005	4,725	4,611	4,310	17,651
12	22	213.04	213.50	214.74	216.86	4,042	4,742	4,596	4,290	17,670
13	25	241.89	242.14	229.47	235.80	5,868	7,404	4,715	5,570	23,557
14	25	248.03	248.08	246.03	250.84	6,557	7,729	6,389	6,850	27,525
15	25	244.06	244.60	245.50	247.37	5,844	7,338	6,305	6,530	26,017
16	28.5	274.57	274.04	274.86	276.74	9,222	9,961	9,491	9,600	38,274
17	28.5	278.09	277.59	279.76	277.10	9,686	10,170	9,580	9,620	39,056
18	28.5	277.28	276.38	278.17	274.30	9,658	10,046	9,923	9,300	38,927
19	30	306.40	308.47	308.78	306.06	13,150	13,971	13,747	13,020	53,888
20	30	307.10	308.47	307.92	306.48	13,084	13,526	13,357	13,000	52,967
21	30	308.12	306.89	309.85	307.43	13,224	12,978	13,617	13,150	52,969
22	32.5	341.56	345.95	344.79	345.65	17,314	17,365	18,105	17,660	70,444
23	32.5	338.41	337.62	345.73	341.03	16,839	17,239	17,957	17,130	69,165
24	32.5	338.52	333.72	345.09	341.42	16,634	17,040	18,219	17,120	69,013
25	Full	388.21	383.46	382.49	386.28	22,404	24,115	22,515	22,780	91,814
26	Full	388.67	384.86	384.74	390.52	23,570	24,425	22,069	23,350	93,414
27	Full	388.66	384.05	383.93	389.01	23,328	24,262	22,789	23,160	93,539

*S.H.P. for No. 4 shaft taken from No. 1 shaft R.P.M.-S.H.P. curve using R.P.M. of No. 4 shaft.

STEAMING TRIALS.

Speed run	Full power	30 knots	25 knots	20 knots	17.5* knots cruis. engine	15 knots cruis. engine	15 knots main engines	10 knots cruis. engine
Date.....	8/19/23	8/18/23	8/16/23	8/15/23	8/14/23	8/15/23	8/14/23	8/14/23
Duration, hours	4	4	6	4	2	4	2	4
R. P. M. mean.	385.43	292.11	235.79	185.78	161.26	137.96	138.11	90.07
Knots corresponding....	34.87	30.01	25.02	20.00	17.46	15.07	15.07	10.01
Shaft horsepower.....	94,920	46,808	23,340	11,355	6,765	4,523	4,595	1,101
No. boilers in use.....	12	12	6	4	3	3	3	3
Avge. press. at boilers:								
F. R. No. 1..	272	255	256	257	...	252	...	...
F. R. No. 2..	276	258	...	...	...	...	...	...
F. R. No. 3..	273	265	...	...	...	...	...	...
F. R. No. 4..	264	260	261	261	260	266	263	257
Avge, air press. in F. R. ins.:								
F. R. No. 1..	7.5	3.7	4.6	5.6	...	3.5	...	...
F. R. No. 2..	7.8	5.8	...	...	...	...	...	...
F. R. No. 3..	7.9	4.3	...	...	...	...	...	...
F. R. No. 4..	7.9	6.0	4.2	5.5	5.0	4.0	4.0	4.6
Press. at turbine throttle for'd. E. R...	247	248	245	246	254	244	253	255
Press. at turbine throttle after E. R...	243	249	250	248	249	253	251	250
Vacuum, ins. mercury for'd E. R.:								
Starboard...	27.6	28.5	28.5	28.4	29.0	28.9	28.9	29.0
Port.....	28.3	29.0	29.0	29.0	29.0	29.0	29.0	28.4
After E. R.:								
Starboard...	28.6	28.5	28.0	28.0	28.1	28.5	28.0	28.5
Port.....	27.4	29.6	29.3	29.3	29.4	29.7	29.5	29.1
Temp. injection water, Deg. F.:								
For'd. E. R..	48	50	50	50	52	51	52	50
After E. R..	52	52	50	54	53	54	53	52

*Live steam admitted to H.P. turbine.

AUXILIARY MACHINERY.

Auxiliary machinery in use	Full power	30 knots	25 knots	20 knots	17.5 knots cruis. engine	15 knots cruis. engine	15 knots main engines	10 knots cruis. engine
Date	8/19/23	8/18/23	8/16/23	8/15/23	8/14/23	8/15/23	8/14/23	8/14/23
Duration, hours	4	4	6	4	2	4	2	4
FIREROOMS								
Boilers	12	12	6	4	3	3	3	3
Blowers	12	8	4	4	2	3	2	1
Burners	72	37	21	14	9	9	9	7
Fuel oil pumps	8	4	2	2	1	2	1	1
Fuel oil heaters	8	5	2	2	1	2	1	1
Booster pumps	2	2	2	1	0	1	1	1
Auxiliary feed pumps	4	0	0	0	0	0	0	0
Feed water heaters	4	4	2	2	1	2	1	1
ENGINEROOMS								
Main feed pumps	4	4	4	3	1	1	1	1
Condensate pumps	5	5	5	4	5	5	4	3
Lubricating oil pumps	5	5	5	5	5	5	5	5
Fire and bilge pumps	2	2	1	2	2	2	2	2
Oil cooler circulating pumps	2	2	2	2	2	2	2	2
Auxiliary condensers	2	1	1	1	1	1	1	0
Air ejectors	...	4	4	4	...	4	4	4
Circulating pumps	0	0	0	0	2	0	1	4
50 K. W. generators	0	1	1	1	1	1	1	0
100 K. W. generators	2	1	1	1	1	1	1	1

The standardization trials were started on 11 August, 1923, on the Vashon Island, Washington, measured mile course and were not finished until 13 August, 1923, due to fog and smoke from forest fires obscuring the range beacons. Thereafter, the trials were continued through without interruption.

ENGINEERING PERFORMANCES AND THE RULES FOR ENGINEERING COMPETITION, U. S. NAVY.*

By Commander C. H. J. Keppler, U. S. Navy, Assistant Director of Fleet Training, Navy Department.

INTRODUCTION.

A man of war, in order to keep her engineering personnel and material in the highest state of efficiency and readiness for war, must be able to operate under any condition with reliability and economy. For the purpose of comparing the relative engineering efficiencies of combatant ships as established by their general performances throughout the year, the competitive system was inaugurated in the Navy and the "Rules for Engineering Performances" gradually developed.

It is not intended to dwell upon the Rules themselves, but rather to attempt an explanation of the reasons why the Rules exist, why they contain the provisions they do, what features of efficient engineering operation each provision intends to bring out, and why they are modified from time to time in order to keep pace with new ideas and the latest installations.

As one of the functions of the Office of Fleet Training, *i.e.*, the correlation and analysis of reports,—these remarks represent a summary of ideas contained in the numerous reports, criticisms and recommendations submitted by individuals or organizations from every part of the service. That the enthusiasm and the spirit for continued development and increased efficiency in operation and maintenance of the machinery equipment of all classes of ships are being promoted, can definitely be accepted from the numerous critical reports and constructive suggestions that are being continually offered by all the Forces afloat.

*This article is in substance the same as a lecture delivered by Commander Keppler recently before the Post Graduate School at the U. S, Naval Academy.

HISTORICAL.

The Departmental supervision of target practice in the U. S. Navy was inaugurated on October 24, 1901, when an officer was assigned by the Bureau of Navigation to duty as Inspector of Target Practice, under that Bureau. From that time on there was a gradual development along gunnery lines, and in 1903 competitions amongst ships in gunnery excellence was established. The engineering competition feature was not included until after the cruise of the Battleship Fleet around the world in 1908 and 1909. During this cruise the necessity for a recorded comparison of engineering performances of vessels was made apparent by extraordinary economies effected through the establishment of a so-called fuel economy competition. Thereafter the preparation of the record, and analysis of engineering performances was added to the duties of the office of the Inspector of Target Practice. In November, 1909, the cognizance of the Inspector of Target Practice was taken from the Bureau of Navigation and established as an office under the Aide for Operations. In February, 1911, the name was changed to the Office of Target Practice and Engineering Competitions. In this same year Submarines were first included in the competition, the gunnery and engineering merits were combined as a comparison of battle efficiency, and the first battle efficiency pennant was awarded to the U. S. S. *Michigan*. In April, 1916, the name of the office was changed to that of Gunnery Exercises and Engineering Performances. Today in accordance with G. O. No. 108, dated 6 June, 1923, the office is known as the Division of Fleet Training and exists for the purpose of assisting the Chief of Naval Operations in the direction of tactical matters, maneuvers, exercises and the training of the Fleet for war. Quoting parts of paragraph 4 of G. O. No. 108, the Division of Fleet Training is charged with:—

(*a*) "The preparation of general instructions for the conduct of fleet exercises.

(*b*) "The preparation of the War Instructions, and other standard instructions, manuals, and drill books governing the military activities of the Fleet in war and in training for war.

(*c*) "The collection, analysis, and review of all data in regard to fleet training and compilation of the same into suitable reports for the information and guidance of the service."

To this end the Director of Fleet Training is expected to administer the office, to examine reports of battle inspections, to keep the Chief of Naval Operations informed as to the readiness of vessels for war service, insofar as pertains to their engineering efficiency and the use of their armament, and to make timely and suitable recommendations with a view to increasing their battle preparedness. The duty of the Assistant Director for Engineering Performances is to carry on all technical work necessary in the collection, analysis, review and dissemination of useful data regarding the engineering performances of all classes of ships. From the annual analysis of performances by the Office of Fleet Training, the detailed instructions for exercises and practices for the next year are prepared. Carefully edited reports and instructions, based on an analysis of the past year's performances, must therefore contain the consensus of service opinion, and by informing all officers of the latest accepted practices, must produce continued advancement towards efficiency by making permanent all good practices, and by preventing waste of time and energy in attempts to investigate methods that have already proved to be without value.

IMPORTANCE OF THE ENGINEERING PERFORMANCES.

The importance from a military point of view of administrating the gunnery, aircraft, and torpedo activities of the Navy, of coordinating effort towards offensive development of naval craft of all types, of increasing steaming efficiency and promoting economy in engineering practices and of keeping the Chief of Naval Operations informed as to the readiness of the Navy for war, is sufficiently apparent to need no

comment. The work of the office is also important from an economic point of view, in that its standardization of methods and coordination of effort by units of the Fleet, result in accomplishments of greatest value to the service in minimum time; and while carrying out assigned programs, increased economy in the expenditure of material and supplies. For example, "the interest in economy has reduced the size of the Navy's fuel bill, has permitted the ships to do more steaming, and has increased the steaming radius of the battle fleet."

It is of first importance that the fleets shall have their engineering training as nearly as practicable under conditions that will develop and train the personnel, and test or improve material for use in war; and the competitive feature must be incorporated whenever it does not produce artificiality, since competitions promote the spirit of rivalry,—which is bound to foster new ideas,—insure improvement in design of machinery, stimulate interest in the study of engineering subjects, and develop a continuous effort towards excelling the accomplishments of others who are engaged in the same activities, with similar equipment.

The application of a competition in engineering covering all performances both underway and in port, is most conducive to the standardization of practice along the most approved lines, to the detection of material defects or faults of personnel, and to the assurance that prescribed tests and inspections are being carried out. It compels constant attention upon the engineering plant of a ship, its condition, its amount and method of operation, its efficiency and its reliability. Efficiency in all other activities and phases of the naval profession may avail nothing if efficiency in the operation and upkeep of the engineering plant is lacking, and the competitions serve to impress upon all hands the essential importance of having the engineering plant in all our ships, at all times, ready to meet any demands that war might bring upon them. Good engineering practice, even where no competition applies, demands that leaky condensers, broken-down pumps, leaky or

salted boilers, hot bearings or similar derangements, be repaired and conditions rectified at once. "Visible leaks are visible evidence of careless engineering."

Engineering performances differ from gunnery exercises in that the competition is a continuous one from one year's end to the other. The engineering competition begins with the first day of July each year and from then on the engineering exercises are a day-to-day necessity, and very often receive recognition only when something goes wrong. If the ship breaks down underway the engineering department is likely to be condemned and engineering comes into unfavorable prominence for the time being. If the ship is run economically and efficiently all the time, that fact is taken as a matter of course and without any special recognition. There is very little that is spectacular in the engineering competition, no flash and roar connected with it, and this fact must be recognized in order to whole heartedly support the spirit of the Rules for Engineering Performances.

The engineering performances comprise a very necessary function in the development of Fleet efficiency. It must not be forgotten, however, that they are only a factor, and as such must be properly controlled, and their limits defined. Engineering performances are directly concerned with:—

(*a*) the training of personnel,

(*b*) promoting economy (I'll have more to say on this word "economy" later).

(*c*) operation and management of the engineering plant.

(d) upkeep and maintenance of the engineering equipment of ships.

They are indirectly concerned with logistics insofar as the fuel estimates necessary for the movement and operation of fleets or forces are concerned; and also, with the design and construction of the engineering installations of ships, and were instituted to obtain efficiency, economy, and standardization of engineering practices and methods. The Rules for Engineering Performances are designed to:—

(*a*) stimulate interest in engineering by comparison of results obtained from the reports of all ships in the same class.

(*b*) measure results by recording similar data under standard conditions on all ships.

In accomplishing these purposes they are also intended to coordinate the efforts of the personnel and to keep in the minds of all hands the necessity for always keeping the ship in the best condition to meet the contingencies of war service.

Fleet efficiency results from individual ships' efficiencies; and ship efficiency in turn includes engineering as one factor. Ship efficiency must take precedence over its engineering efficiency, and for that reason there are certain principles which the Rules must recognize, namely;—

(*a*) First: Rules should furnish no motive, directly or indirectly, for operating under any particular condition, but on the contrary should measure the results obtained in whatever condition or activity is desired by competent authority.

For instance, the present Rules are not supposed to reward any operating condition at the expense of any other condition. In the first sets of Rules, that is those published up to about 1917, fuel expenditures could be juggled so that expenditures could be taken away from factors that were unfavorable, and added to factors under which higher multiples could be made. The present Rules frustrate this system by the simple means of having the final score dependent upon the ratio of the *total* fuel allowed to the *total* fuel used, so that all expenditures appear eventually as *fuel used* no matter whether underway, at anchor, or standing by, or any other condition. In case any one condition is favored by trying to apportion the fuel to another more favorable condition, besides being of no benefit in the final score, this manipulation is reflected in the next year's standard allowances by having them reduced, since any year's allowances are based principally on the best previous actual performances as reported.

(*b*) Second: The comparison established by the Rules should be based on results from the same installations, or in

other words upon "Sister ships," or vessels with installations so similar as to permit their being grouped together. Any comparison for competitive standing between different installations, while desirable for improvement in design, is undesirable on account of its possible inaccuracy as a measure of personnel efficiency. With identical or similar installations, the results are an accurate measure of personnel efficiency. Therefore, the comparison should be based on results obtained now and the average results obtained previously with the same installation over a considerable period, say three to five years. This is the principle on which the assignments of new standard fuel allowances, as is shown later, are established.

(*c*) Third: The Rules should consider the ship as a unit. The efficiency of the evaporating, refrigerating or electrical plants; or while standing by, raising steam, etc., and the considerate use of fresh water, heat and electricity, are all components that concern the ship as whole and collectively determine the engineering efficiency of the ship. Therefore, in order to obtain the highest degree of efficiency and economy, *complete cooperation between the deck and engineering departments* is essential at all times.

To sum up, competitive engineering performances were instituted for the general improvement of Engineering in the Navy, for the stimulation of interest in engineering throughout the personnel, and for the elimination of waste in the use of fuel, oil, water and supplies. However, the Rules must be so drawn that these results may be accomplished without jeopardizing a ship's chances in the competition, and still enable her to supply such amounts of light, heat, fresh water or ice as are essential to the comfort, health and happiness of the crew. Restriction or curtailment of heat, light, ventilation, fresh water, running boats, etc., are contrary to the spirit and intent of the Rules and are not considered economy. The large gains in economy are within the engineering plant itself, and unless there be a serious shortage of fuel, the Rules expressly forbid such restrictions. As a measure of the routine en-

gineering performances, the present Rules, which have resulted from the recommendations of engineer officers and others interested in engineering progress of the Navy during a period of many years, are considered to be as simple and to serve their purpose as well as any that could be devised under the existing great range of operating conditions, military activities, and diversity of types.

It is believed that in cases where the engineering personnel are not familiar or interested in the Rules covering the competitive features of the engineering work, the fault lies more in the neglect to properly instruct such personnel rather than in the ambiguity of the Rules. To promote efficiency in engineering and arouse interest in the competition it is necessary that the intelligent understanding of the enlisted personnel be secured.

The crew of a gun does not, as a rule, work out its final score at target practice, but from the number of hits made in the target, can form a good idea as to whether or not a creditable record was made. In the same manner, if proper graphical methods are employed, the engineer's force can tell, by the number of buckets of coal or gallons of oil used, under the various operating conditions, whether or not effective results are being attained. In other words, it requires education. Lectures or talks should be used to instruct the personnel of the engineering department in the Rules of the game, and the reasons for the various Rules should be explained to them. "Blackboard talks" or discussions which relate to the work in hand or contain understandable data on the previous day's work will create enthusiasm and keep up the interest of All Hands. There should be graphs showing the gallons of fuel oil or buckets of coal allowed for a score of 100 for each of the different conditions of operation, and a curve of actual performance should be drawn and kept up to date alongside the 100 curve so that the personnel can tell each day, or each watch, whether or not they are doing creditable work, by a simple comparison of the curve of actual performances with the stand-

ard graph for the existing conditions. Similar "pictures" should also be prepared, and kept up to date, showing satisfactory performances in the use of electricity, water and lubricating oil.

RULES AND ALLOWANCES.

Particular care was exercised in preparing the Rules for the current competition, to correct the standard allowances in accordance with actual performances and the numerous acceptable suggestions that were received from time to time, and it is believed that the present allowances are more nearly practicable and more nearly represent attainable performances by any vessel in each class or group, than ever before. This appears to be evidenced by the fact that for the first six months of the current engineering year 67 destroyers out of the 109 in commission have final scores between 90 and 112; and 13 out of the 16 eligible battleships are between 90 and 117.

Furthermore, the present practice of assigning standard fuel allowances determined from actual best performances of previous years, results in an accurate and actual comparison for each year of the vessel's performances, and also gives a comparison of the efficiency of the personnel in the use of the same installation.

Also the inclusion in the competition of awards of Navy "Es" and money prizes to the ship of each class that shows the greatest improvement in engineering over the previous year,—performances for both years being referred to a common standard,—has been an excellent measure of comparative engineering efficiency of the personnel. The year 1921-22 furnished a striking example of this feature when the battleship that received this award improved her score about 28 per cent, with no alterations or additions to her installation but through the changes in administration, operation and upkeep, concurrent with an almost complete change of her personnel shortly before the beginning of the year in which the noticeable improvement was made.

In order that the data available from the monthly Engineering Performance Record, "Form H," may be more useful for logistic purposes, and also that the relation between Engineering Performances and Logistics may be continuous and more intimate a set of fuel allowances to be known as Logistic Fuel Allowances is being developed for all competing ships. The development of this phase of Engineering Performances it is hoped will overcome one of its criticisms, *i.e.,* that due to the annual change in standard fuel allowances the competition did not, without extensive and laborious computations, afford a ready means of comparing the actual performances of a vessel through a series of years, and also that the information available on the Form H was not immediately useful for logistic purposes because of the annual change in the standard allowances and because the data were not separated into performance "underway" and performance—"not underway." The new Form H provides this information at a glance, and enables a vessel to compete continuously with her own previous performances as well as with the performances of other vessels.

These logistic allowances are based on the actual performances over several years, for the battleships the last five years were taken and the allowance computed for each year which would have given a score of 100 underway, the average of these five values was then taken as the logistic allowance. This procedure was worked out for 7 different speeds, *i.e.,* 4, 8, 10, 12, 15, 18 and 20 knots, and it is believed the resulting data are exceedingly accurate and should permit close predictions of fuel consumptions. In the future therefore, in addition to the standard fuel allowances, vessels will be assigned a factor, to be known as the Logistic Conversion Factor, by which the standard allowances for that year are to be multiplied in order to give the logistic allowance. This conversion factor may change from year to year accordingly as it becomes necessary to change the standard allowances in order to maintain an equitable comparison of a ship's engineering performances for competitive purposes and awards. For any one vessel,

unless of course there is a radical change in her machinery installation, the logistic fuel allowances will not change from year to year and are expected to,—

(*a*) Afford a ready means of comparing a vessel's, a division's or a fleet's performance for one year with that of any other year.

(*b*) Refer all logistic computations in regard to fuel to the same basis, and

(*c*) Present the fuel consumption data in such a form as to be a maximum usefulness in the planning of Fleet maneuvers and the making of annual fuel estimates.

In the destroyer class numerous conflicting recommendations were received from different sources, varying from a flat allowance,—as low as the best performances of the best ships,—for all ships, to special requests from individual ships for allowances even greater than had ever been assigned. With so many vessels of the same class in service it is manifestly impossible to assign individual allowances covering every possible operating contingency, and still retain the competition within limits where all factors can be equitably controlled, accurately checked, and correctly co-ordinated by the reviewing facilities available in one office. Accordingly it is necessary to assign common standards to groups of ships whose characteristics are sufficiently alike to insure that the common standards will be fair to all in the group under the average operating conditions that obtain over long periods such as the competition year. It is admittedly possible that the comparative performances of some one ship may be temporarily adversely affected by reasons of some peculiar installation, or unusual activity not required of all vessels in the group, but since the performances of each ship are carefully scrutinized and her actual performances under each of the different operating conditions given careful consideration, in connection with the performances of the best ships with similar installations, it is believed that the differences in the steaming conditions for any unfavorable periods will be very nearly equalized by the

general law of averages that will be involved in a year's activities.

As the condition of "operating without boilers" has now become a well recognized engineering practice, a change from existing requirements was necessary to provide for this method of operation. The previous provision of the Rules whereby the supplying ship received as an allowance the fuel used during such periods, and consequently a score of 100, was undesirable for several reasons; the principal ones being that it removed the competitive feature, defeated the spirit of the Rules, and annulled the necessity for All Hands to observe the strictest economy or to carry out efficient engineering practices. Accordingly, a percentage allowance based on the standard hourly at anchor allowance and the number of vessels supplied, was established for destroyers and light mine layers. In order to discourage any tendency to restrict heat, light, water, etc., no credit is given for supplying more than three vessels at a time.

There were strong recommendations from various sources to eliminate certain fuel expenditures such as fuel used in steam launches, testing boilers or machinery after overhauls or repairs, smoke screen work, or in special maneuvers which all vessels of a group did not happen to perform. The recommendations, however, were generally disapproved as the Office of Fleet Training is opposed to the principle of eliminating any fuel expenditures incurred in the routine and regular performances demanded of ships. A special allowance for smoke screen work has been granted, however, in order to encourage this form of training and to compensate for the reduced efficiency due to fouling of boiler firesides and uneconomical fuel expenditures incurred while producing smoke screens.

In several instances where ships received derangement penalties, due to breakdown of various machinery units wherein no member of the engineer's force happened to be actually concerned at the time, such as steering gear casualties, strong arguments were advanced that the penalty should be waived owing

to the responsibility involved. However, attention is invited to the fact that the spirit and intent of the Rules for Engineering Performances do not contemplate that penalties should be assigned only for casualties wherein some member of the engineer's force is involved, but recognize that all casualties to material, under the cognizance of the Engineer Officer, are subject to the standard penalty, and that the personal element includes any member of a ship's complement. The Rules contemplate prize money awards to any member of the crew of a trophy winning ship, and, conversely, it follows that any member of the crew may also incur responsibility leading to a penalty award for an engineering material casualty in which he may have been concerned. As noted in the Rules, they are also intended "to stimulate the efforts of the personnel and to keep in the minds of all the necessity for the strictest attention to duty." It is desired to emphasize the fact that the engineering competition is a ship affair, not restricted wholly to the engineering department, and that any officer or man on board may contribute to the success or failure of the ship.

The engineering competition is intended to impress upon the operating personnel the necessity at all times for keeping their material installations in the highest state of readiness and perfection. The remission of penalties for material casualties would relieve the personnel of such necessity and also of the idea that the final merit in the competition must include every factor that affects their relative efficiency in comparison with every other vessel in their class. The elimination of penalties for material casualties would also remove the incentive to prove to the material bureaus that certain installations possess inherent weaknesses, and also tend to modify the pressure that the operating personnel will continue to bring on the responsible heads for corrective measures. Rigid adherence to penalty awards for material breakdowns will impress upon "all hands" the fact that if the material under their cognizance can not perform its designed functions then it is incumbent upon them to make every effort to get such material changed or improved.

At one time in the engineering competition it was the practice to reduce all speeds to a standard speed for each class of vessels. Because of the widely different operating conditions, and possible inaccuracies of the reduction formula, it was found that this resulted in inequitable comparison of performances. The present practice allows a certain amount of fuel for each tenth of a knot speed, each allowance being based on best past performances, and the vessel's merit is dependent upon her approach to this allowance. The present "conditions," that is, operation underway, at anchor, periods at Navy Yard, standing by, etc., have been found necessary in order to avoid inequitable and unjust comparisons. There have been, and will continue to be, routine military activities required of ships which necessitate "standing by," or "light spread," conditions for days at a time. The occupation of Vera Cruz or the typhoon season in Manila are cases one can remember where ships have been required to stand by for weeks, and at such times those ships are justly entitled to extra fuel allowances which a "standing by" condition provides.

Any material change in the present "conditions" should only be made after full consideration of its possible effect on the equity of the final comparison, which the scores establish, and of the possibility of obtaining erroneous logistic data.

The argument has been advanced by numerous ships on various occasions, that, under certain conditions or with special combinations of machinery, a vessel could obtain a higher score if greater than necessary fuel expenditures were incurred. This is the inevitable consequence of the impossibility to devise absolutely and theoretically correct Rules and allowances to cover every possible contingency. The best that can be done is to attempt to assign standard allowances, based on all available information, that, as far as possible, will give correct comparisons for the average and ordinary conditions which apply during the very great majority of the time, on the assumption that gains or losses, due to unusual circumstances will neutralize each other over extended periods. In recog-

nizing this feature of the principle on which the allowances must necessarily be based, the office of Fleet Training has felt confident that officers would not incur unnecessary expenditures of fuel for the sake of improving the engineering score, and has taken the opportunity of so expressing itself on several occasions in correspondence with ships, as an announcement of the convictions of the office as to the principles that will guide officers in these special cases that must unfortunately sometimes arise.

DETERMINATION OF STANDARD FUEL ALLOWANCES.

In determining the standard fuel allowances, at anchor for instance, the actual past performances of each ship are used, together with the latest average performances of the class or group of ships having similar installations, neglecting any that show excess expenditures beyond what are known to be within a vessel's attainment.

In every case consideration is given to the average performances of similar vessels in the class, and of course, also where the vessel concerned individually shows a performance radically different from her class, due consideration is given to either decreasing or increasing her allowances in order to assign a standard with which she should be able to equitably compete with the other ships in the class. This procedure is possible of course in the office of Fleet Training where the data for all vessels of a class are available and under consideration, and is a feature that the individual ships often lose sight of when criticising their allowances. In other words, the office tries to maintain a "Bird's Eye View" of the whole subject and to avoid partiality to any one ship or class.

Similar processes are involved in establishing allowances for the other operating conditions of "Standing By, Light Spread, Raising Steam," etc. For the current year all Navy Yard Performances have been eliminated from the calculation of a vessel's merit. This change has met with universal approval owing to the widely different conditions that prevail

regarding weather, locality, and the magnitude of repair work that is being undertaken. The length of time in a Navy Yard, however, is still one of the eligibility requirements, *i.e.*, more than 180 days out of the year disqualifies any vessel for awards.

In establishing allowances underway, the steaming data sheets or curves of actual performances are kept plotted up to date, and the new fuel standards for the various speeds determined from the curve plotted through the spots of most recent best performances. Then, for "Sister Ships" or vessels with installations so similar as to permit their being grouped together, the best performanee curve of each is plotted on another sheet and a resultant obtained which serves as the basic curve for the group from which the standard allowances, after considering certain modifications for individual ships whose consistent performances vary materially, are made.

As the standard fuel allowance assigned for periods underway depends ultimately upon the speed revolution curve the ship is using, it is essential that all vessels possess an accurate clean bottom standardization curve. This curve should be made as soon as possible after docking and the instructions laid down in the Rules for Engineering Performances regarding the preparation of the standardization curve strictly followed. At present, the Board of Inspection and Survey is conducting a series of trials with a destroyer in order to determine the effect upon the speed-revolution performances of a ship because of differences in displacement and the effect of fouling of the bottom due to varying lengths of time out of dock. The *Tennessee* on the West Coast is also engaged in similar trials. The November, 1923, number of the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS has a very good article giving the first results of some of these trials held during the past year and a half by the *Brooks*. The very unexpected fact is developed that for a constant number of revolutions at certain points,—*i.e.*, at the lower speeds,—increased displacement gave increased speeds. This fact was

repeated in subsequent trials of the one of the new light cruisers, and the *Tennessee* also reports that on recent standardization trials she made the same speed, about 18.90 knots, on 3 R. P. M. less, with nearly a thousand tons greater displacement, than she did on her first standardization trials. An interesting case, showing the variations that will creep in and affect the fuel consumption data of sister ships under practically identical conditions, occurred on the trip of the *Arkansas* and *Wyoming* from Puget Sound to New York in the summer of 1921. They were docked within a few days of each other, coaled at the same time with the same kind of coal, left the yard and steamed together in formation, held exactly the same maneuvers under identical conditions of sea and weather. From Puget Sound to Panama the *Wyoming* was flagship and guide, from Panama to New York the *Arkansas* was flagship and guide, yet upon arrival at New York the records showed that the *Arkansas* had steamed 301 miles more than the *Wyoming* and had averaged nearly a half knot more speed per hour. This difference was the resultant effect of all differences in draft and trim of the two ships, their steering, throttle keeping, propeller slip, and the inaccurate speed-revolution curves. This last item probably was the controlling factor and produced a constant error in the same direction, while the others very likely neutralized each other, but the results of this voyage show how essential are correct standardization curves, since it is seldom that two sister ships' long distance steaming together gives such exact comparative data for showing up discrepancies in their speed-revolution curves.

Many ships, and destroyers generally, expend at sea more than their fuel allowances. There is undoubtedly big room for improvement in operation underway and it has been recommended from several sources that the best way to bring this about is by requiring that two economy trials be held, and that no vessel be permitted to conduct full power trials until successfully completing the two economy trials. This proposition was submitted to all forces afloat last year and numerous

recommendations, some of them conflicting, were received. At a meeting of all Fleet and Force engineers, while the Fleet was in Panama, which considered the many acceptable suggestions, it was the consensus of opinion that the compulsory holding of economy runs preliminary to full power trials should not be made a part of the Rules; but that if existing provisions of the Rules were faithfully carried out in regard to the daily analyses, little could be gained, and the additional paper work and complications involved were not desirable. It was believed that the holding of so-called operating trials, as already instituted in some commands, accomplishes the desired results; *and that every run was an Economy Run,* and that further progress lies along lines of simplification of existing rules, a conservative attitude toward increasing the number of extra allowances, and an intensive study of daily performances, both underway and at anchor, to eliminate errors in operation and to determine the source of losses.

THE DAILY ANALYSIS.

In order to promote economy and obtain information upon the material conditions and actual operating capabilities of the Engineering Department a system of daily analysis of engineering performances has been devised. "This striking the daily balance," is for the purpose of enabling the engineer to keep in close touch with the operation of the plant and provide a means for the early detection of faults or poor performances. The daily analysis must be closely coupled with active and thorough supervision. "It is everlasting thoroughness that insures economy and not an occasional fine performance." It matters not how thoroughly operating methods have been worked out, if their execution is not constantly checked they will lapse into a careless performance. By this careful check it is possible to quickly locate the cause of the increased fuel expenditure before it runs into a matter of hours or days and the time element greatly increases the loss." The analysis for any piece of machinery is based upon balancing the theoretical

or guaranteed performance against its actual performance. The absence of measuring devices such as steam flow meters or measuring tanks renders impracticable an analysis that would be possible in a laboratory, but sufficient data can be obtained from the water rates of the various pieces of machinery, and the rate of evaporation of each boiler, so that comparisons from day to day will give the relative performance of any unit; and a poor performance or the falling off in efficiency can be detected by these daily comparisons. Striking the daily balance consists of multiplying the hours in operation for each piece of machinery by the horsepower it develops and its guaranteed water rate, and adding the products for all machinery in operation during the day. This will give a figure in terms of pounds of water representing the amount of steam the plant should have used. The difference between this figure and that obtained by multiplying the number of pounds of fuel actually used during the day by the rate of evaporation of the boilers (expressed in pounds of water evaporated per pound of fuel) is indicative of the economy or a measure of the degree of efficiency which is being obtained.

All the data necessary to make a plant analysis can be taken from the engine room log if it is properly and carefully kept. The main factors are discharge pressures, R. P. M., or strokes per minute, gallons of water distilled, and K. W. H. generated. In the "Naval Institute" for July, 1921, and January, 1924, there are very good articles on daily analysis, and engineering competitions in general.

Right here it is desired to emphasize very strongly this idea of the daily analysis, it is the greatest individual factor towards fuel economy in any efficient administration of an engineering department. It is not a mathematical puzzle as most people seem to think, but is the only true and successful method of finding out where your fuel goes. Every power plant on shore of any consequence, where daily efficiency and economy is judged in dollars and cents for profits and dividends, includes critical daily examination of the input and output of every

unit in the plant. Ships can likewise profit by pursuing this same policy, and the records of the office show that those ships which have followed a scheme of plant analysis are the ones which stand near the top or show consistently high performances. Hand in hand with the daily analysis goes the planning of the overhaul and upkeep work necessary in every engineering installation. Normally most of this work is done at anchor, it is essential therefore that, before every period when not underway, a comprehensive plan of work should be laid out; in other words, make an estimate of the situation, and arrange the schedule of repairs so as to get the greatest possible productive use of the man-hours available, and at the same time figure on the least expenditure of fuel. Here again comes coordination with other departments on board ship, for instance plan work that requires compressed air or air tools on the same day that torpedoes are going to be overhauled or gas ejecting systems tested, do not overhaul generators or their auxiliaries when considerable electric power will be needed on deck, nor test out steering and anchor engines during peak loads of gunnery drills, or while forcing evaporators to catch up on fresh water. In other words without a plan of work, the procedure is spasmodic, there is a lack of system, an unnecessary expenditure of fuel, and the efforts of the personnel do not produce maximum results. The adoption of and adherence to a systematic plan of work, while at anchor, produced such effective results in one of the destroyer squadrons that fuel consumptions were reduced to nearly one quarter their original amounts, and the following is quoted from the report: "It was first necessary to thoroughly clean the fire and watersides of boilers, and overhaul all auxiliary machinery so as to operate the units at the proper slow speed necessary for auxiliary purposes. In order to do this preparatory work, and in all subsequent operation, a system or a 'plan of work' was followed.

"Burning fuel oil and burning it economically was one of the biggest problems because the personnel were adverse to

paying the proper attention to operation, and steady even running of machinery, but after a complete cycle of the routine cleaning and overhauling of boilers and machinery, the crews and officers were convinced that the systematic method was correct. A competitive condition was ever kept before the crews so that they would take proper interest in their work in order to do as well or better than a ship alongside had done the week before."

Defects of operation are generally plainly visible, but their results and losses are not so impressive until they are expressed in numbers. Plant analysis, boiler tests, and planning-the-work eliminate "hit and miss" economy. They answer the question, "where does the fuel go?"

SUBMARINES.

Heretofore, the basis for computing the merit of submarines in engineering performances, was to assign weights to different factors. Each factor covered certain standard requirements which had to be met by the submarines during the competition year. The sum of the weights of all factors was equal to 100, and the merit for the year of any submarine was the sum of the scores made under the several factors.

This year a radical change was made, the competition following the general lines of surface craft. In general a standard allowance of fuel is assigned each vessel, which, compared with the amount used, forms the basis of computing the score. Standard requirements for the full power trials, availability trials, and hours of operation, surface and submerged, are assigned, and failure to meet these requirements results in a penalty to the ship's score. Unlike surface vessels which are given variable fuel allowances based on a fuel expenditure-speed curve, submarines are given one standard fuel allowance per mile for both engines running and another allowance per mile for one engine running. This was done because the variations in the speeds of submarines are so small and the average speed for the year of vessels of the same class almost

identical, and also, inasmuch as the fuel expenditures reported vary so greatly on different runs, while the best performances vary so little over a wide range of speeds, no fuel expenditure speed curve can be plotted which would put all vessels of a class on an equal competitive basis.

The battery efficiency of submarines is measured by a comparison of K. W. H. input and output. A standard fuel allowance being given per K. W. H. of discharge and another for K. W.H. of charge.

When establishing the present system one of the primary considerations was to make the method of reports as simple as possible, as it was realized that submarines should not be overburdened with complicated administrative paper work.

It is felt that the present system is an improvement over former years and that as time goes on all the difficulties will be ironed out and the submarine competition will receive the same enthusiastic support and be on the same equitable basis as the competition now is for surface craft.

DISADVANTAGES AND CRITICISMS OF THE ENGINEERING COMPETITION.

The general criticism of the competitive idea in engineering and gunnery exercises has usually been that undue efforts have been exerted to enlist competition as an element for promoting efficiency and cooperation. When we push competition too far and try to make the Rules of a game the means of governing so that a man or ship may get rewards through winning in the game rather than in improving the game, we drift away from the real spirit and intent of the game. The temptation "to beat the Rules" of the competition without striving for real efficiency may lead to artificiality and to the undermining of honest endeavor. The fears raised by all these objections, however, are believed to be unwarranted when we consider the positive benefits to the training and education of the personnel, and to the care and operation of the material, that have resulted during the 12 or 15 years competitions have existed,

and furthermore when we admit the convictions, we must all possess, that officers and men are guided by principles of loyalty, honesty and the desire to "play the game fair."

Another disadvantage pointed out is that in order for all competitors to have confidence in the justice of the result it is necessary to measure the comparative efficiency of the units by numerous absolute standards. These standards may be wrong through errors in computation or they may not fit every conceivable operating condition or type of installation, hence individuals believe they should be entitled to special and extra allowances. This procedure if carried out becomes full of intangibles and the use of many erratic factors arises. It also necessitates complicated Rules and a cumbersome system, and involves so much administrative mechanism that the energy of the organization is largely consumed in the operation of the mechanism rather than in the accomplishment of results contributing to the original object in view. The tendency of the Engineering performance Rules has been towards a reduction of the number of factors, and the determination of standings or comparative efficiencies upon the general and average conditions or factors that obtain over such long periods as a competition year.

SOME ADVANTAGES AND RESULTS OF ENGINEERING COMPETITIONS.

Aside from the incentive to achieve the highest economy of fuel consumption, which is one of the principles upon which the Rules are based, the acute shortage of funds for fuel during the past two or three years acted as a further stimulus in this direction. Also the longer periods at anchor gave better opportunities for overhaul and upkeep of machinery, which improvement in the condition of the plant is reflected in the reduced expenditures of fuel, especially at anchor, which nearly all vessels effected.

(*a*) The recent past performances of many of the battleships have attained figures that only a few years ago would

have been considered impossible. From a generally accepted expenditure of from 18 to 24 tons of fuel a day at anchor, many ships now operate on from 12 to 15 tons, while a daily performance of between 8 and 10 tons is not at all unusual. Such performances are only possible by reason of the concentrated and conscientious efforts of every officer and man on board, by intensive training and instruction of the personnel on the most approved and scientific methods of burning fuel, by the strictest adherence to the excellent instructive orders and bulletins on economical and efficient engineering methods promulgated from time to time by the Bureau of Engineering, by constant attention to the maintenance and upkeep of every unit in the machinery installation,—with the idea of obtaining results approaching as nearly the designed performances as is possible,—and by the careful analysis of all performances over regular periods and the application of immediate corrective measures to any part of the engineering plant that shows deterioration or inefficient performance. Of course, it is needless to say that improvement in design of equipment and replacements with modern installations, by the material Bureaus in recent years also has contributed largely to the results attained.

(*b*) Likewise many destroyers show individual at anchor performances approaching only one-half their original expenditures, and vessels steaming for from 3 to 5 additional boats have maintained daily averages of much less than they previously expended for themselves alone. The consumption of the steaming vessel when supplying steam and power to several others has gradually been reduced, as better facilities for connecting up with each other for steam, water or electricity have been acquired. Also after the idea had become firmly established, the spirit of cooperation and desire on the part of every vessel of the group to attain the greatest economy was awakened. The results of the methods employed are evidenced in the reduction of fuel expenditures during a six-months' period, accomplished by one destroyer squadron, the one that had a

"plan of work," referred to above, where within a period of six months the average daily fuel expenditures were reduced to one quarter their previous consumptions.

Aside from the material reduction of fuel expenditures that such performances produce, there is the enormous advantage to the "dead ships" of having steam off their machinery so that repairs, inspections and overhauls can be more easily and thoroughly accomplished, and furthermore that all the artificer complement on board is available for repair work and the services of none lost because of watch standing.

(*c*) The engineering performance game stimulates new designs or modifications in existing types of machinery, which service demands have proved are deficient in performance or defective in operation. For instance, recently the material bureaus have taken energetic steps towards redesign of the steering engines of certain classes of ships. Owing to their numerous breakdowns these vessels incurred derangement penalties which materially reduced their engineering scores, and led to vigorous protests with consequent action by the responsible material Bureaus.

(*d*) The full power trial requirements of the Rules have led to exhaustive investigations concerning effects on speed and horsepower of differences in displacement and varying lengths of time out of dock. The standardization trials of the *Putnam* and *Brooks*, the *Tennessee* and the new light cruisers have resulted in much valuable information that may change some of our preconceived ideas of design of hull and propellers, and which now appear to warrant the expense of considerable time and money in continued investigation, to determine why, for instance, in all three of the above types at certain points, a decided increase in displacement resulted in no decrease in speed for constant revolutions, and at the lower range of speeds an actual increase in speed took place.

(*e*) Enthusiasm towards minimum fuel consumption in port has made the nesting together of several destroyers, with only one steaming from the group, a well established practice and produced a marked saving of fuel.

(*f*) This same idea stimulated efforts towards the perfection of a new type of burner for oil boilers combining blowers and burners in single units. Also there have been developed a number of electrically driven auxiliaries for port use, and all capital ships will be equipped with these auxiliaries.

(*g*) The auxiliary and miscellaneous vessels since they have been required to submit the regular Engineering Performance Records, while not assigned fuel standards nor given competitive standings, have nevertheless become keenly interested in economical performances and made noticeable reductions in their fuel consumptions and improvements in their methods of operation.

(*h*) From only a preliminary inspection of their performance reports numerous cases of individual ships making poor performances, in comparison to their sister types, have been uncovered. The engineering departments of such ships have been investigated or inspected and corrective measures applied.

It is believed that the above gratifying results are due primarily to the policy of engineer officers exacting strict compliance with the Department's instructions regarding methods of operation, routine inspections, and tests. Engineer officers nowadays must give very close supervision to the management of their departments in order to insure that the above mentioned instructions are being carried out. It has been demonstrated while operating with the modern installation, that the old, experienced, enlisted personnel will not change the methods of operation they learned years ago, for modern methods, until practically *forced* to do so by the engineer officer. In order to establish methods in accordance with the latest approved engineering practices, an engineer officer must be thoroughly familiar with all instructions concerning the operation and management of his plant and also to have sufficient self-assurance to override the objections of his long experienced assistants when their method does not coincide with the modern practices.

CONCLUSION.

In conclusion, let us consider for a moment, just what is meant by the word "economy." The office of Fleet Training has long been impressed with the failure, on the part of the service, to grasp the pure meaning of this word. Apparently it believes that economy means parsimony, stinginess and a general miserly attitude. The word in its pure sense means *efficient administration, i.e., prevention of waste.* It does not mean absence of spending, but it does mean spending wisely and to the best advantage. Very often military necessities demand generous expenditures of fuel or supplies, but the successful accomplishment of these military demands is not attained unless these expenditures are wisely and efficiently utilized. The Rules suffer because of the lack of understanding of the word "economy," and in spite of the tremendous good that they have accomplished they are condemned by thoughtless people because of lack of understanding, and because of failure on the part of their opponents afloat to become imbued with and to carry out the spirit of the Rules. Wholehearted support of the principles of the engineering competition by all the personnel throughout the service, and more particularly an active interest by those in the higher commands, are also essential to success, because these will lend encouragement and support towards economical, progressive and satisfactory engineering performances. Diligent study and application of the latest standard practices, careful analysis of performances of all units of the engineering plant, and patient, faithful training and instruction of the personnel will all contribute to progress, to economy, to efficient operation, to improved material conditions and to higher scores. All of these are factors that will help maintain the Fleet in the highest state of battle readiness.

INDIVIDUAL FORCED DRAFT FUEL OIL BURNERS.

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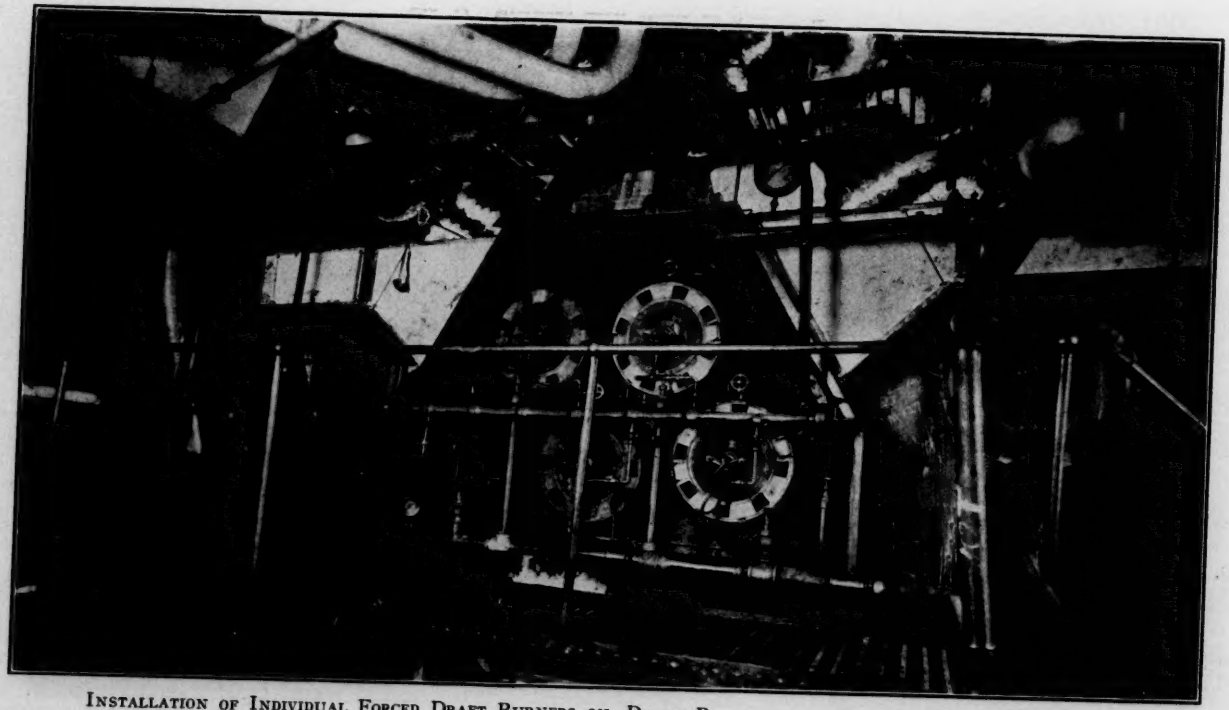
Navy Department.

In order to burn oil under a boiler it is necessary to supply air to support combustion. The amount of air necessary to burn a pound of ordinary fuel oil varies from 180 to 200 cubic feet. This is the amount necessary for complete combustion but to insure that complete combustion is obtained excess air in various amounts must be supplied. If the percentage of excess air is kept below 50 the results may be considered very good, as experimental tests where the most careful attention is paid to combustion show 20 per cent excess air to be an excellent figure.

The air necessary for combustion of the oil may be supplied in two ways; by natural draft or mechanical draft. Mechanical draft may be produced (*a*) by a steam jet in the base of the stack; (*b*) by suction on the stack side of the boiler (induced draft) or (*c*) by pressure on the furnace (forced draft). Method (*c*) only is in use on naval vessels where mechanical draft is used.

Draft in a boiler is the difference in pressure between the furnace and the fireroom available for producing a flow of the gases. If the gases within a boiler and stack be heated, they will expand and so have a less density than the outside air. Therefore, the weight of the column of hot gases is less than the weight of a column of cold air of the same height. The unit pressure in the furnace is therefore less than in the fireroom. This difference in unit pressure will cause a flow of air into the furnace and up the stack. This draft is called



INSTALLATION OF INDIVIDUAL FORCED DRAFT BURNERS ON DYSON BOILER AT THE FUEL OIL TESTING PLANT

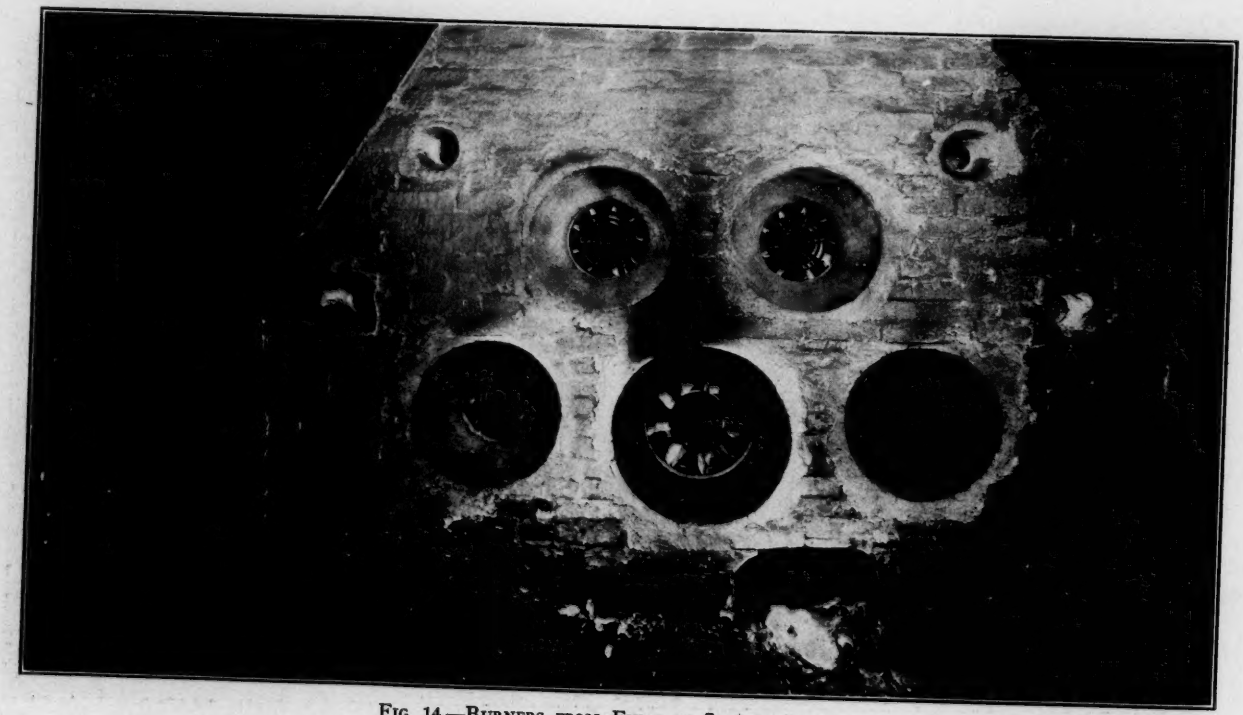


FIG. 14.—BURNERS FROM FURNACE SIDE OF BOILER.

natural draft and its intensity is usually measured in inches of water. It is measured thus because the pressures obtained are usually too small to be expressed in pounds (one inch of water corresponds roughly to .036 pound per square inch).

From the above definition of draft the following formula may be worked out:

$$D = .52\ P\ H \left(\frac{1}{T} - \frac{1}{T'}\right) \text{where}$$

D = draft in inches of water
P = atmospheric pressure in pounds per square inch
H = height of stack in feet
T = absolute atmospheric temperature
T′ = absolute temperature of stack gases.

In this formula the density of the stack gases is taken to be the same as that of air at the same temperature.

For various stack temperatures, assuming atmospheric temperature and pressure as 60 degrees F. and 14.7 pounds respectively, the formula reduces to D = KH in which the values of K for various stack temperatures are:

Temp. stack gases	750	650	550	450	350
Value of K	.0084	.0078	.0071	.0063	.0053

This is the theoretical draft which can never be obtained in practice because of draft losses. These losses are:

(*a*) Loss in stack and uptakes due to friction and inertia of the gases;

(*b*) Loss in the boiler due to the arrangement of the tubes and baffles;

(*c*) Loss due to accumulations of soot on tubes;

(*d*) Loss due to cooling effect of air leakage.

The draft loss due to friction in the stack may be expressed by the formula:

$$d = \frac{f W^2 C H}{A^3}$$

where

d = draft loss in inches of water
W = weight of gases passing per second pounds
C = circumference of stack, feet
H = height of stack in feet
A = area of passage square feet
f = constant (.0015 at 600 degrees F., .0011 at 300 degrees F. for steel stacks).

The stack loss may be evaluated quite closely from this formula. The other losses, however, are indeterminate and, therefore, the draft cannot be computed exactly.

It is found in practice that with temperatures ordinarily encountered, a handy rule for determining natural draft may be stated as follows:

The actual draft intensity in the furnace expressed in hundredths of an inch of water will be approximately one-half the vertical distance in feet from the point of measurement to the top of the stack. That is, a stack 100 feet high will give half an inch draft.

It is easy to see why very little oil may be burned with natural draft in a Naval boiler, particularly a destroyer boiler. The stack height of a destroyer above the burners is about 43 feet. Assuming the stack temperature in port to be 550 degrees F., the theoretical draft is about .3 inch. Tests have shown, however, that actually the draft in the furnace is about .2 inch.

The area through the Fore River conical type register is 41.65 square inches. The velocity of air produced by .2 inch draft is 1,790 feet per minute. The theoretical flow of air through the register is 518 cubic feet per minute but due to the resistance of the air register, the actual flow is probably nearer 360 cubic feet per minute.

This is enough air to burn efficiently about 1½ pounds of oil per minute or 12 gallons per hour.

With the same draft a Bureau conical register will allow enough air to go through to burn efficiently 22 gallons per hour. The Bureau natural draft register has an area of 111 square inches and so will allow enough air through to burn 32 gallons an hour. These amounts of course can be somewhat increased by the use of windsails—one destroyer was able to burn 42 gallons an hour with Bureau natural draft registers but the combustion was not efficient. The practice was always dangerous due to frequent periods when the draft was not sufficient to provide air for the combustion of the oil being pumped in through the atomizers. The accumulation of oil on the boiler floor resulted in flarebacks and explosions. This fault of natural draft operation on destroyers was appreciated and the use of forced draft blowers is now mandatory unless a vessel is fitted with natural draft burners only.

The total air capacity of the forced draft blowers on our vessels has been divided into as small a number of units as is consistent with sound engineering practice, having in view the remaining possible development of power after one or more units are disabled. Thus on destroyers with four boilers in two firerooms, six forced draft blowers are installed, three in each fireroom. On the *New Mexico* with nine boilers in three firerooms, nine blowers are installed, three in each fireroom, and on the *Tennessee* with eight boilers in eight firerooms one blower is installed in each fireroom. For full power conditions, all blowers are run practically at their limiting safe speeds.

The necessity for large amounts of air at high pressure when vessels are developing full power results in very disadvantageous blower conditions when vessels are at cruising speeds with a power demand of from 5 to 15 per cent of full power, and in tremendous losses under port conditions, when as little as ½ per cent of the oil burned at full power is consumed, due to the fact that it is manifestly impossible to

install a sufficient number of forced draft blowers to insure that those in use will be operating at maximum efficiency at all times.

The desirability of supplying air for combustion by other means than the large forced draft blowers, particularly when at anchor, was recognized soon after the introduction of oil for fuel, but until recently none of the proposed methods have been satisfactory from a practical point of view.

Experiments were made to cut down the amount of air delivered to the fire room by putting plates in the downcomers to the blowers and in the fans themselves. A small auxiliary steam line under boiler pressure was led direct to the steam belt of the turbine so that the losses attendant on the use of reduced pressures would be eliminated.

On other vessels, the fireroom was put under the pressure needed for port oil burning by mounting an electric driven fan in the hatch leading down to the fireroom.

Some destroyers enclosed the register in an air tight casing and connected the casing to the discharge of a 1,000 cubic foot ventilating blower electrically driven.

Other ships mounted portable blowers at the fireroom hatches and discharged air through windsails close to the burners.

While many of these schemes functioned satisfactorily on the vessels originating them, all were makeshift arrangements and were not considered safe enough from an operating point of view to make them standard installations.

In 1922, the Fuel Oil Testing Plant in charge of Lieut. Comdr. H. H. Norton, U.S.N., developed a port auxiliary burner. This burner was made up of a Coppus turbo-blower bolted to the boiler front and an oil burner proper which was led in to the side of the fan casing with a goose-neck so that the oil was discharged to the furnace in the usual way. The fan was 12 inches in diameter and was designed to run at

3,000 R.P.M. At this speed, it would deliver 2,500 cubic feet per minute at ¼ inch static pressure or 1,050 cubic feet per minute at 1¼ inches static pressure.

Tests were run on this unit at the Fuel Oil Testing Plant at various rates of combustion and very successful results obtained. A Yarrow boiler of 4,725 square feet of heating surface was used.

Duration of test hours.....	3½	3½	3	3	2	4½	2
Flue gas CO_2 per cent.....	7.6	11.6	13.4	13.7	13.2	14.6	14.9
Evaporation F&A/pounds oil....	14.98	15.38	15.36	15.42	15.17	14.93	15.42
Pounds of oil per hour.....	189	381	522	632	715	756	867
Oil/square foot H. S.	.04	.08	.12	.14	.16	.17	.19
Oil gallons/hour.....	25	50	70	83	95	100	114
Boiler and Furnace Eff.....	78	80.1	79.7	80.2	77.8	79	80.3
Turbine Steam Consumption in per cent of steam made.....	..	1.49	1.45	1.60	1.95	1.89	2.25*

These tests were so encouraging that a similar unit was installed on the U. S. S. *Williamson* in the summer of 1922. The burner pipe, however, was led through the shaft of the turbine, which was made hollow to accommodate the burner pipe. It was intended to use auxiliary exhaust pressure to run the turbine but the fluctuations in this line were so marked that steady operation of the blower was impracticable and resort was had to live steam for operating the turbine. This burner has been used successfully in port and at sea since

*The exhaust from the turbines is condensed after passing through the auxiliary exhaust line.

installation with a great saving in oil. Using a 1/16 inch plug and tip, steam is generated for the following auxiliaries:

1 Auxiliary feed pump
1 Fuel Oil pump
1 Fire and bilge pump
1 Auxiliary air and circulating pump
1 Generator carrying the lighting, ventilating and ice machine load.
Evaporating plant operating on auxiliary exhaust.

With the above auxiliaries in use the daily fuel oil consumption averages about 575 gallons against 750 to 800 gallons with a regular burner and forced draft blower.

When using this burner, it is unnecessary to put any pressure on the fireroom. All hatches and blower intakes are open and the air necessary for combustion is forced into the furnace by the small blower. While this burner was devised primarily as a port burner, its possibilities for use at sea were recognized and tests were projected using a number of these burners on a single boiler.

Additional Coppus turbines with fans of 16 inch and 13 inch diameter were purchased and the burners and furnace castings manufactured at the Navy Yard, Philadelphia. A boiler at the Fuel Oil Testing Plant was fitted completely with these burners and extensive tests were carried on. The frontispiece shows Dyson boiler fitted with these burners (16-inch fans).

These units were made up of a Coppus turbo-blower attached to a furnace casting which held a bladed cone and diffuser for properly directing the flow of air from the fan to the burning oil.

Figs. 1 and 2 show one of the 13 inch complete units attached to a boiler with the shutter open and closed. Fig. 3 is a cross section of the furnace casting showing the bladed cone and diffuser. Fig. 4 is a plan of the diffuser and Fig. 5

FIG. 1.—13-INCH BURNER, COPPUS BLOWER (SHUTTER OPEN).

FIG. 2.—13-INCH BURNER, COPPUS BLOWER (SHUTTER CLOSED).

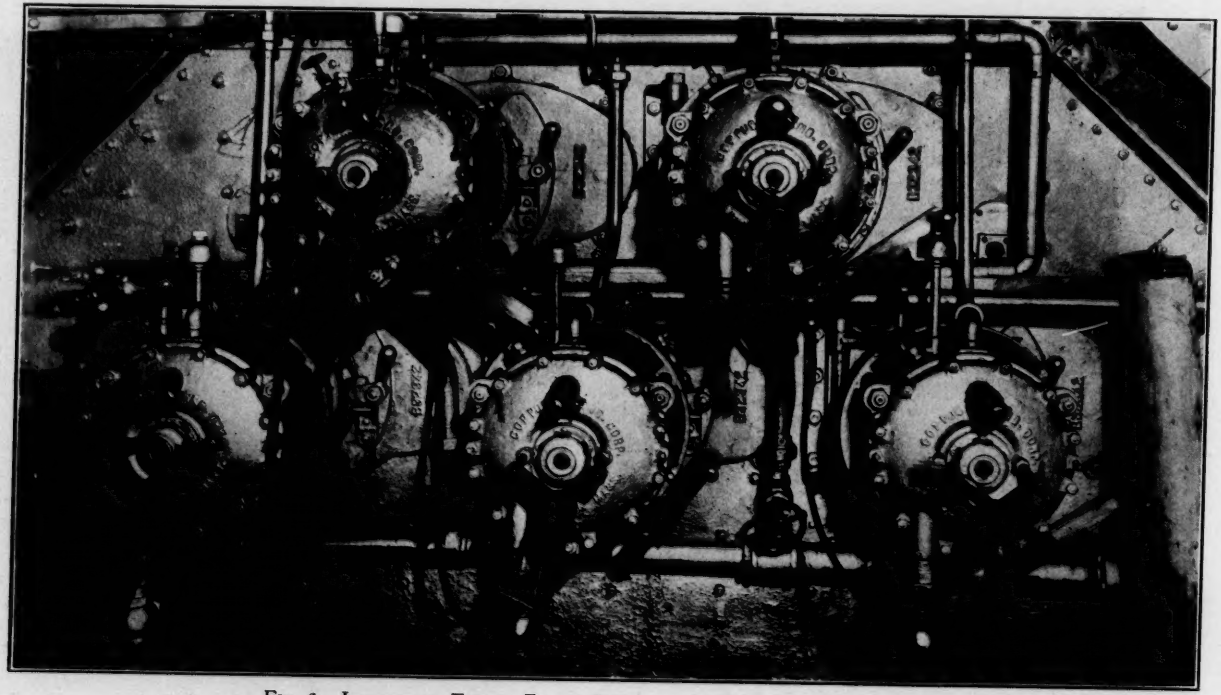


FIG. 6.—INDIVIDUAL FORCED DRAFT BURNERS ON U. S. S. "MAYFLOWER."

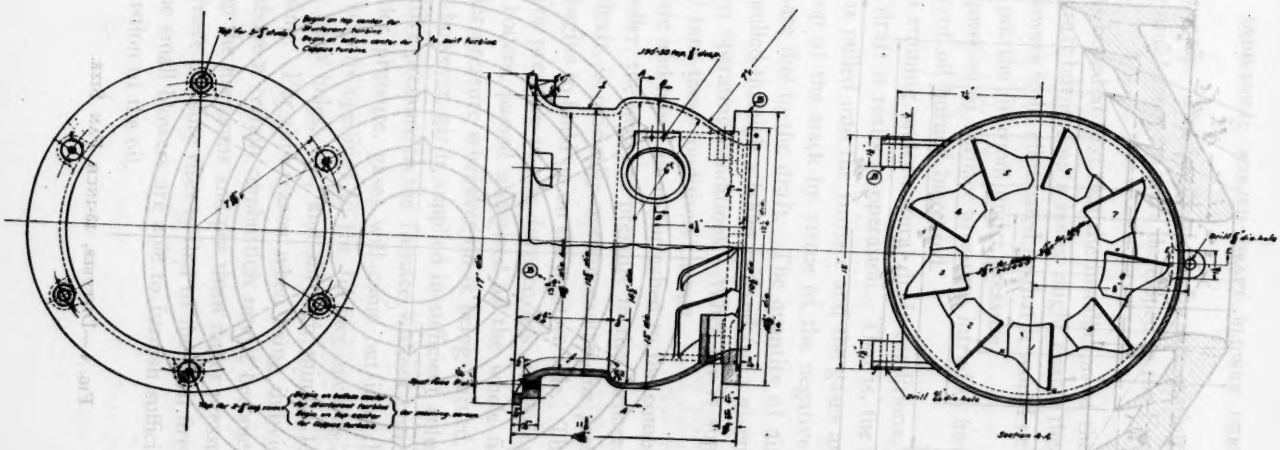


FIG. 3.—FURNACE CASTING, 13-INCH FAN SIZE.

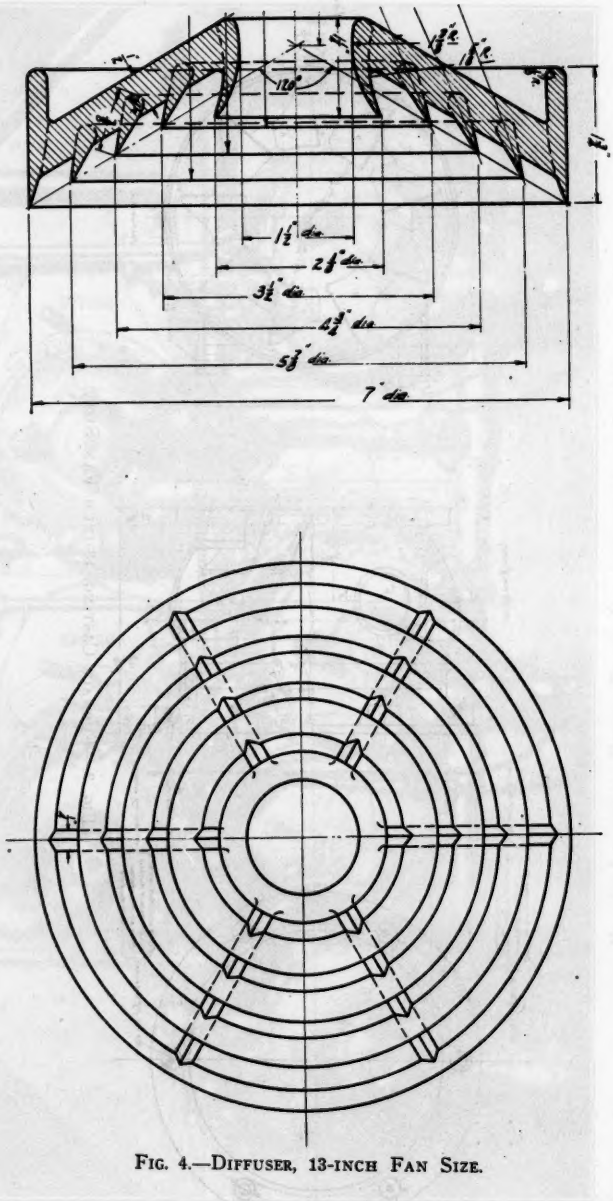


FIG. 4.—DIFFUSER, 13-INCH FAN SIZE.

the burner tip. The details of the furnace casting, bladed cone, diffuser and burner tip were developed at the Fuel Oil Testing Plant.

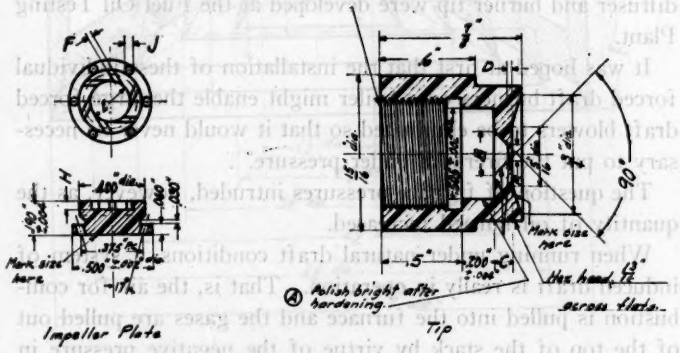
It was hoped at first that the installation of these individual forced draft burners on a boiler might enable the large forced draft blowers to be eliminated so that it would never be necessary to put the fireroom under pressure.

The question of furnace pressures intruded, however, as the quantity of oil burned increased.

When running under natural draft conditions, a system of induced draft is really in operation. That is, the air for combustion is pulled into the furnace and the gases are pulled out of the top of the stack by virtue of the negative pressure in the furnace due to the draft. The quantity of air which will thus be pulled through a boiler is fixed for a given installation and given operating conditions. When air is supplied to the boiler by fans these conditions change.

As more and more air is furnished to the combustion space of the boiler, the negative pressure in the furnace, due to the natural draft, is gradually overcome until the pressure in the furnace begins to build up on the positive side. This condition of positive pressure in the furnace has heretofore, with oil burning boilers, passed unnoticed as the whole fireroom has been under pressure whenever oil is being burned.

When, however, air is supplied in sufficient volume to build up a positive pressure in the furnace without putting the fireroom under pressure, gases will come out into the fireroom through various openings in the furnace such as idle burner registers, peep holes, etc., and through minute leaks in the boiler casing. For this reason when using individual turboblowers above a certain combustion rate it becomes necessary to close up the firerooms and put them under pressure as soon as the furnace pressure rises to that of the open fireroom, even though the small blowers are able to furnish sufficient air for the combustion of the oil.



TIP DIMENSIONS.					
Atomizer Size Mark	Approx. Cap. lbs. per hr. Oil 200 lbs. per sq. in. Gauge	Orifice Dia. A	Dia. Base 45° Cone B	Front Wall Thickness C	Approx. Straight Orifice Length D
4	245 305	1/16	.169	.073	.020
6	540 670	3/32	.236	.091	.025

IMPELLER PLATE DIMENSIONS.							
Atomizer Size Mark	Impeller Plate Size Mark	Dia. of Whirling Chamber E	Slot Width F	No. of Slots G	Slot Depth H	Dia. of Holes in Impeller Plate J	No. of Holes
4	4-15	.174	.030	4	.038	5/64	4
	4-20	.174	.040	4	.038		4
	4-25	.174	.040	4	.048		4
6	6-15	.241	.040	4	.065	5/64	4
	6-20	.241	.040	6	.058		6
	6-30	.241	.040	8	.065		8

FIG. 5.—BURNER TIP FOR BURNERS DEVELOPED AT THE FUEL OIL TESTING PLANT.

From the above discussion, the necessity for the retention of the large forced draft blowers is evident, at least on our present boilers. If a double casing could be provided so that all air for the small blowers was taken from the top of the firerooms, gas leaking from the inner casing would be picked up by the air flow to the fans and sent into the boilers. This scheme would permit the use of positive pressures in the furnace. It is impracticable, however, to put a complete casing around boilers now installed in our vessels.

The point at which it becomes necessary to close up a fireroom and start the large blowers depends on the type of boiler employed, the height of stack (and hence the natural draft) and the amount of oil to be burned.

Tests at the Fuel Oil Testing Plant were made with boilers completely equipped with individual forced draft burners. A test on a B. & W. boiler equipped with four individual forced draft burners showed that a total of 1,500 pounds of oil per hour or 375 pounds per burner per hour could be burned before the pressure in the furnace became positive with a consequent leakage of gas into the fireroom. Similar tests on a Dyson boiler equipped with five burners showed slightly over 600 pounds per hour per burner as the limit with open fireroom operation. The stack height during these tests was 32 feet, giving a natural draft of about .16 inch.

In January, 1923, it was decided to replace the old coal burning boilers of the *Mayflower* with two oil burning boilers built during the war for the *Fanning*. These were Thornycroft boilers with a heating surface of 4,500 square feet each. It was decided if possible to equip these boilers with individual forced draft burners complete and so eliminate the necessity for large forced draft blowers. A test was run at Philadelphia on a B. & W. boiler with a steam jet at the base of the stack to simulate the draft produced by a stack as high as that on the *Mayflower* (.3 inch). Under this condition, using four burners, 556 pounds of oil per burner per hour were burned while still retaining a negative pressure in the furnace.

With this information the Bureau of Engineering ordered the installation of five burners on each boiler. The turbo-blowers for the units were furnished by the Coppus Engineering Corporation while the burners and furnace castings were furnished by the Babcock & Wilcox Company from designs prepared by the Fuel Oil Testing Plant. As can be seen from the photograph Fig. 6 the furnace castings are hinged so that when not in use they can be turned away from the boiler front. At the same time a fire shutter covers the hole in the boiler front and prevents the entrance of air. If work is to be done on the unit a plug can be put into the opening in the brickwork. Fig. 7 shows one of these fire plugs. The turbines were de-

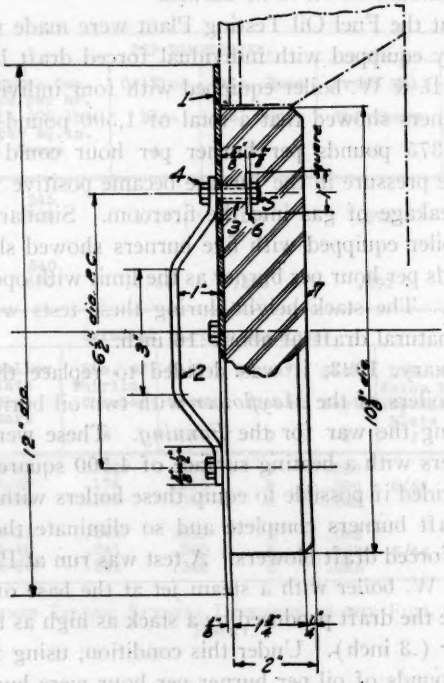


FIG. 7.—FIRE PLUG FOR CLOSING BOILER FRONT WHEN FURNACE CASTING IS SWUNG CLEAR.

signed to run at 4,200 R.P.M. while using steam at 160 pounds pressure and exhausting to atmospheric pressure. At this speed, driving a 13 inch fan, each blower would deliver 3,000 cubic feet of air per minute against 1½ inch static pressure.

The burners were installed at the Navy Yard, Norfolk, in May, 1923. A dock trial was run using the power of one boiler burning about 350 pounds of oil per burner per hour or a total of 1,750 pounds per hour. Later trials underway were run, ending with an hour's full power trial. Making 139.7 R.P.M., which corresponds to 14.7 knots, an average of 477.7 pounds of oil per burner or a total of 4,777 pounds per hour was burned. This was the limit that could be burned with the oil pressure available (265 pounds), and the size of burner tips provided (5 No. 49 B. & W. tips on one boiler and 5 5/64 inch F.O.T.P. tips on the other). During this run, the negative pressure in the furnace was never less than .1 inch and most of the time was .2 inch. This showed that with larger burner tips and a higher oil pressure considerably more oil could be burned. Larger tips have since been supplied and the oil pump turbine nozzles altered so that 300 pounds oil pressure can be put on the burners. On the standardization trials of the *Mayflower*, in July, 1923, a high speed of 15.4 knots was attained. On account of the age of the engines, it was not deemed safe to exceed this speed. The boilers could have furnished much more steam and as the air pressure in the furnaces was still negative, more oil could have been burned. No trouble was experienced, except with the fire shutters, which showed a tendency to stick. These shutters are being replaced with a drum type longitudinal shutter fitting over the fan intake. When the burner is shut down the shutter is pulled to the rear, thus preventing air from entering the boiler. The temperature in the fireroom at no time exceeded 125 degrees F.

The success of this installation on the *Mayflower* decided the Bureau to purchase a number of these blower-burners for use on destroyers and auxiliaries. Contract for 100 of these burners for destroyers and 56 for auxiliary vessels was entered into, the Sturtevant Company furnishing the blowers and the Navy Yard, Philadelphia, making the furnace castings and burners.

The contracts called for 13-inch fans on the turbo-blowers and the burners were to be capable of burning 500 pounds of oil per hour.

The contract for destroyer blowers was later modified to call for 16-inch fans on fourteen of the blowers. Ten of these have been installed on the *Billingsley,* five each on boilers 1 and 2. The installation of burners is similar to that shown in the frontispiece. This installation was made, not only to determine how these burners would function as individual blower-burners under open fireroom conditions, but also how they would operate with the large forced draft blowers running in conjunction with the small blowers, as well as with the small blowers shut down, the burners then functioning as ordinary burners.

This data was desired in order to determine how many burners should be installed on existing vessels, as well as for design purposes on new construction.

The turbines are designed to run at 200 pounds pressure and 10 pounds exhaust, making 3,500 R.P.M. At this speed the 16-inch fans will deliver about 6,000 cubic feet per minute.

The diffuser, bladed cone and atomizer are similar to those installed on the *Mayflower,* except that they are of larger dimensions.

Fig. 8 shows a sectional assembly of the forced draft burner installed on the *Billingsley.*

The burners are arranged on the boiler front in two rows, three in the lower and two in the upper row. Steam for the turbines is supplied through a reducing valve to a steam mani-

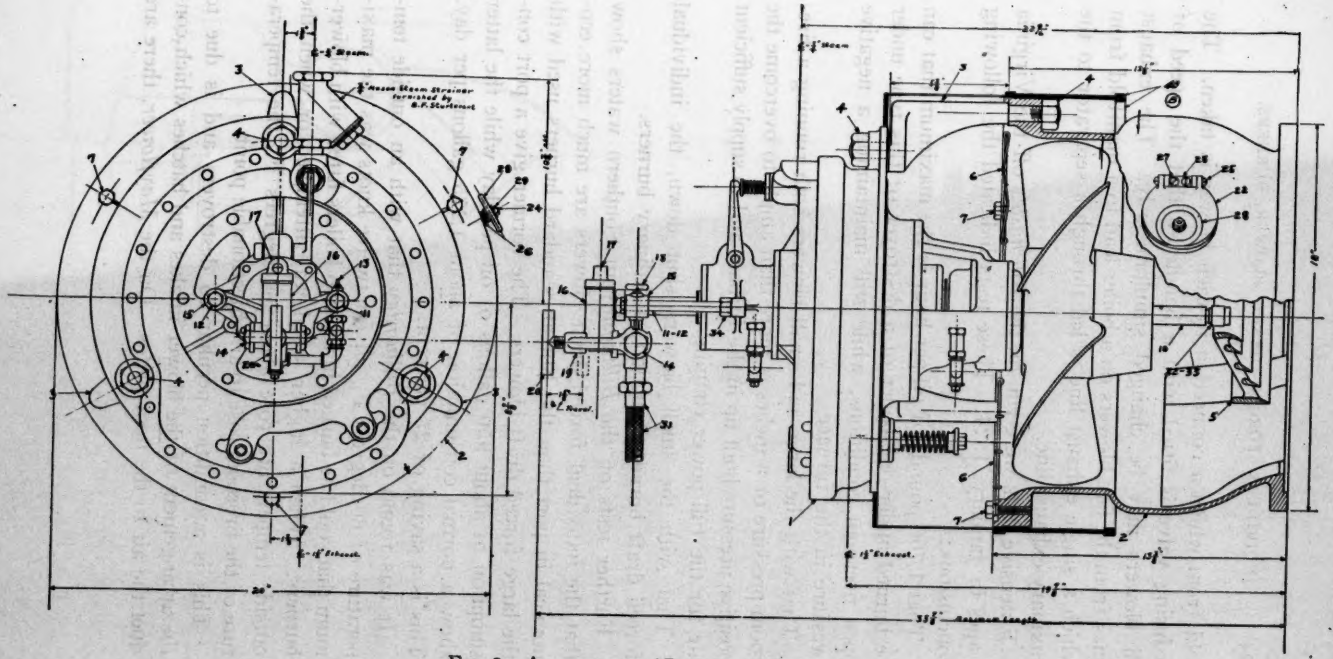


FIG. 8.—ASSEMBLY OF "BILLINGSLEY" BURNER.

fold from which a connection to each blower is taken. The reducing valve is fitted with a handle so that the speed of all blowers may be changed simultaneously. The exhaust lines from the five blowers on a boiler lead to a manifold from which a single exhaust line is led through a separator to the auxiliary exhaust line.

Extensive tests were run on the *Billingsley* off the Virginia Capes in January, 1924. These tests indicated the following conclusions:

That 1,700 pounds of oil per hour is the maximum that can be burned under one boiler of a destroyer of this type under open fireroom conditions, while still maintaining a negative pressure in the furnace.

That with the large and small blowers both running a fireroom pressure of 3 inches is more than ample to overcome the positive pressure built up in the furnace and to supply sufficient air for the full power running.

That with the small blowers shut down, the individual forced draft burners function as ordinary burners.

Further tests on the *Billingsley* in Southern waters show that the individual forced draft blowers are much more economical in port than the Bureau standard burners used with the large forced draft blowers. The former give a port consumption of about 950 gallons of oil per day while the latter show a normal consumption of about 1,275 gallons per day. This is a saving of 25 per cent.

It was found on the *Billingsley* that with an outside temperature of 60 degrees F., oil for twelve knots was the maximum that could be burned in the two boilers fitted with blower-burners. When a higher speed was attempted or when the outside temperature rose above 60 degrees F., the temperature of the fireroom rose to an objectionable point.

This is a condition peculiar to destroyers and is due to the arrangement of the blower intakes and hatches which conduct the air to the floor plates. On the *Mayflower*, there are

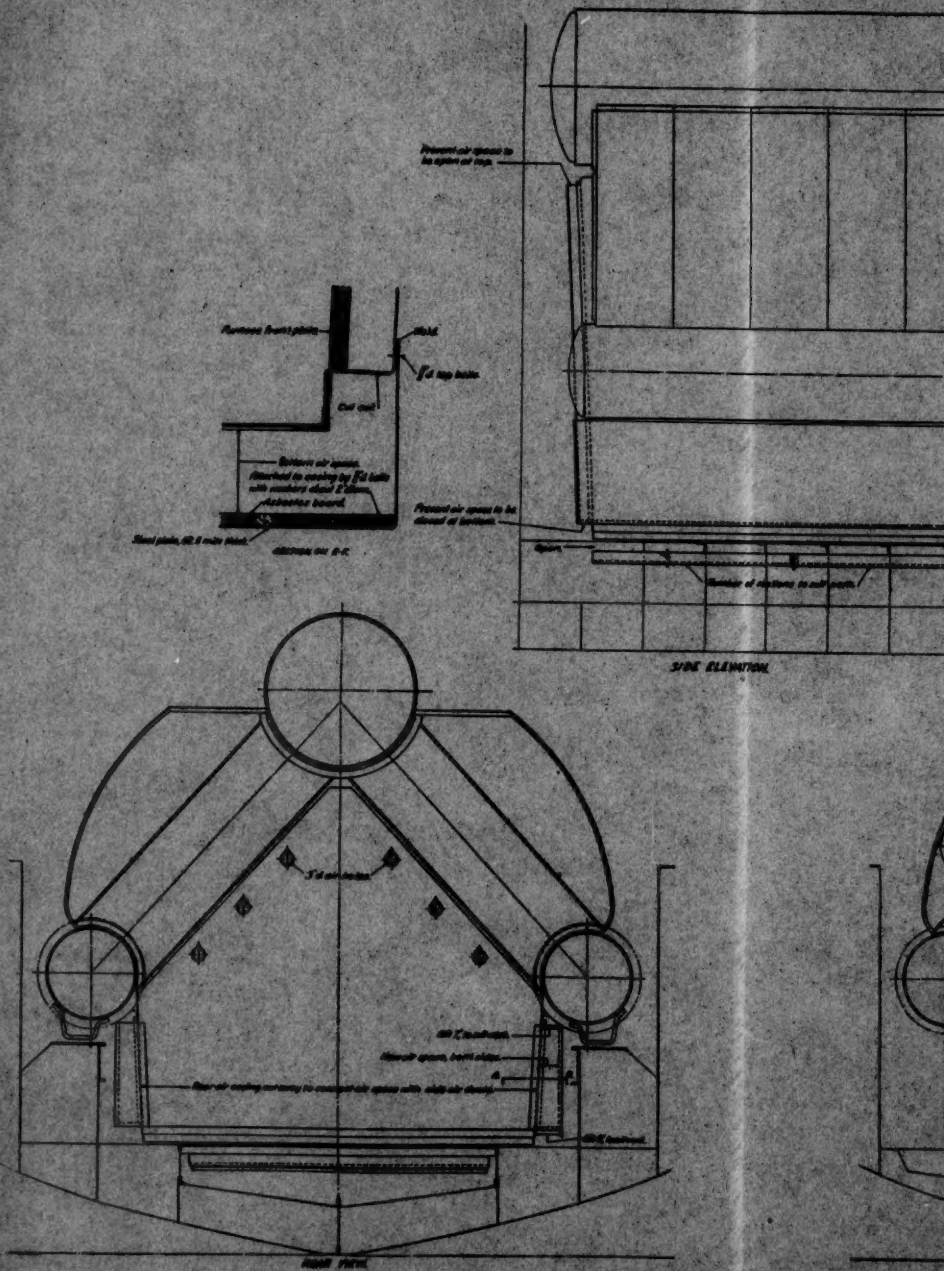
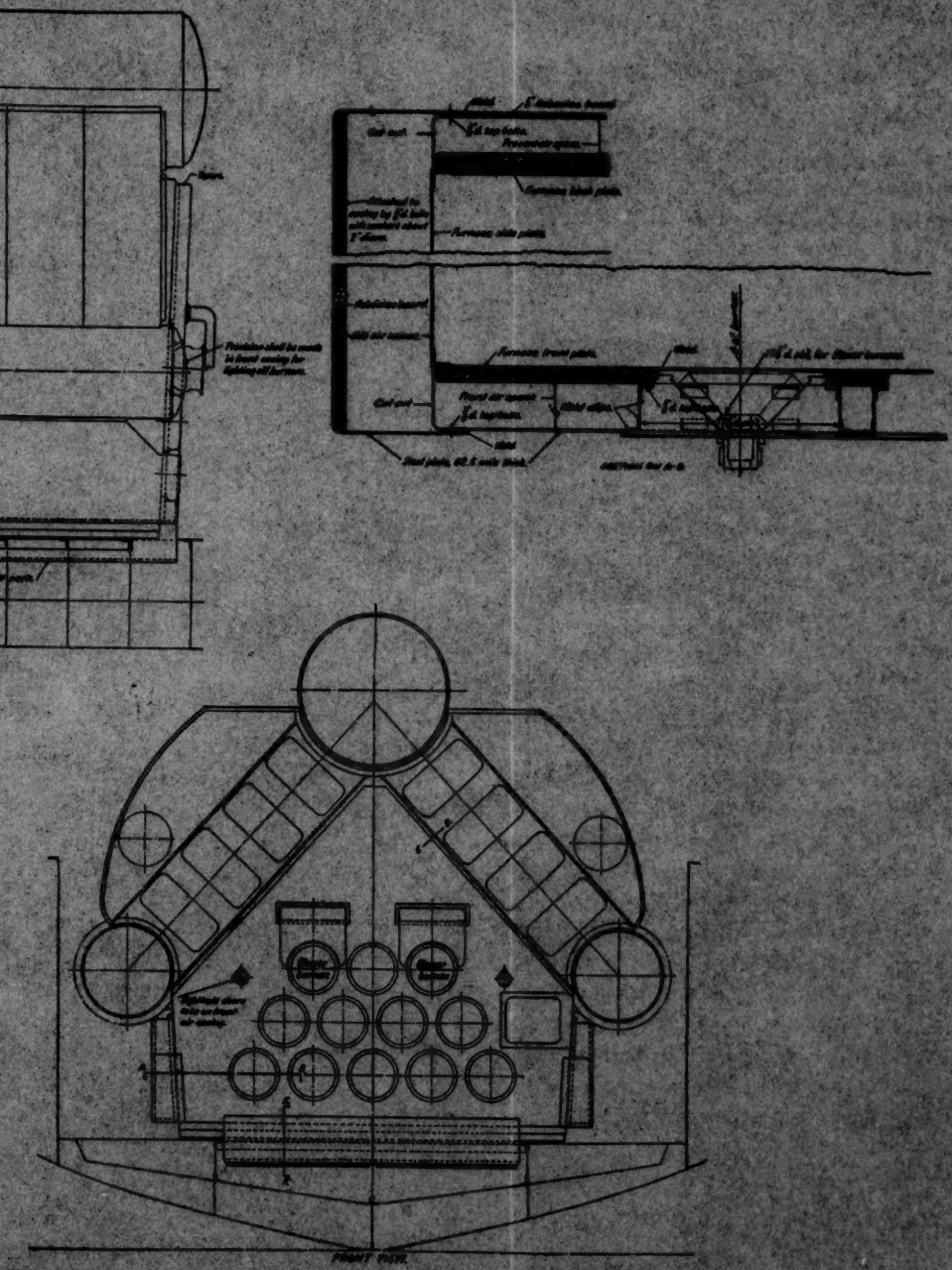


FIG. 9.—BOILER AIR



—Boiler Air Casing.

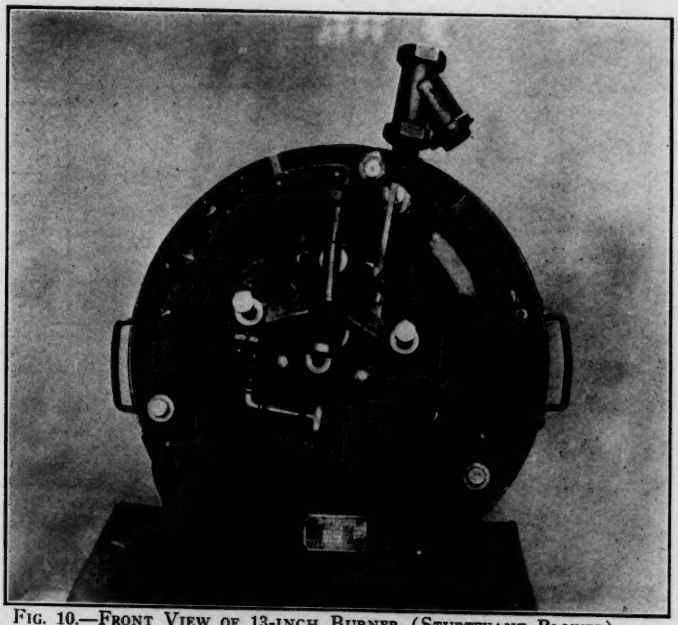


FIG. 10.—FRONT VIEW OF 13-INCH BURNER (STURTEVANT BLOWER).

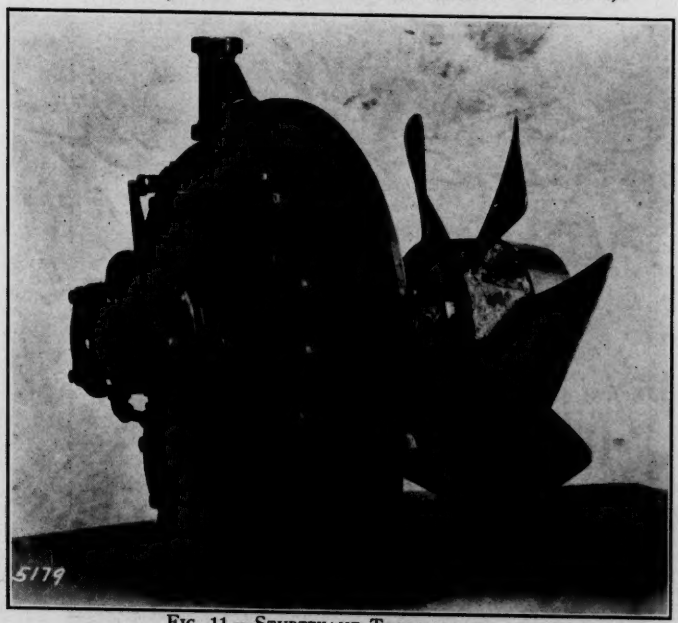


FIG. 11.—STURTEVANT TURBO-BLOWER.

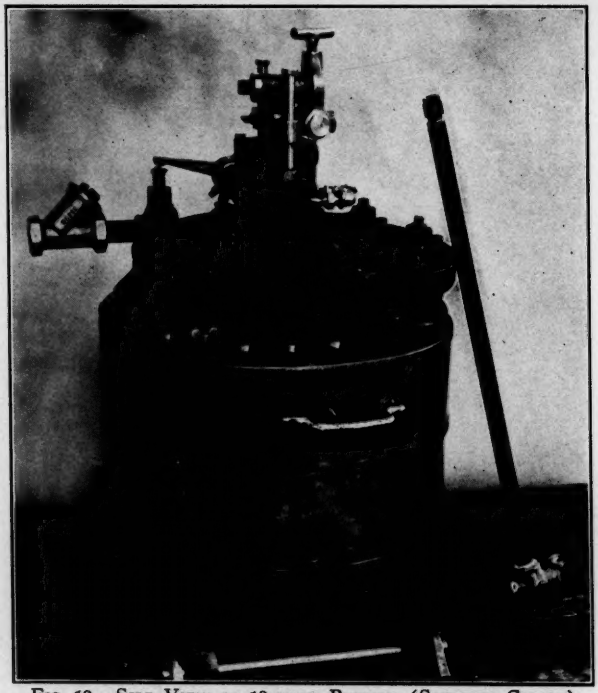


FIG. 12.—SIDE VIEW OF 13-INCH BURNER (SHUTTER CLOSED).

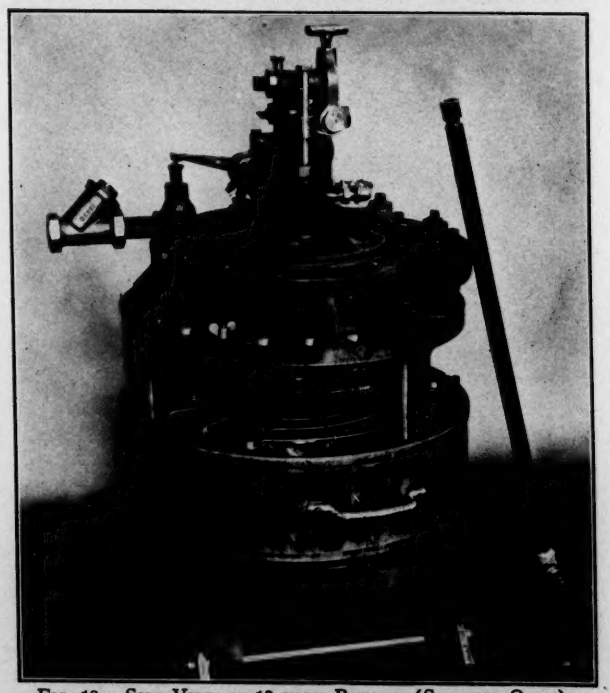


FIG. 13.—SIDE VIEW OF 13-INCH BURNER (SHUTTER OPEN).

two large ventilating ducts which lead directly from deck to the floor plates, and in addition there are ventilating trunks in each corner of the fireroom which act as exhaust trunks. On the *Mayflower,* when burning over twice as much oil in her boilers as did the *Billingsley,* the temperature on the floor plates was never excessive, the maximum reported being 125 degrees F.

The high temperatures found in the open fireroom operation will be corrected in all subsequent installations. As far as possible the brickwork of boilers will be enclosed with insulated casings arranged to give a three inch air space between the present boiler casing and the new casing. These air spaces will be connected to the front of the boiler where the heated air will be sucked into the blower fans and discharged into the furnace; Fig. 9 shows the arrangement as laid out for Yarrow boilers.

These added air casings will reduce the fireroom temperatures to such a point that there will be no discomfort attached to the operation of the individual forced draft burners even in tropical waters.

Fig. 10, 11, 12, 13 show various views of the 13-inch burners.

Fig. 14 shows the burners from inside the boiler furnace.

Fig. 15 is a sectional assembly of the 13-inch burner.

Burners of this type are adapted for use with Scotch boilers and have been authorized for the *Langley* and the *Canopus.* The boilers on both of these vessels will be equipped complete with individual forced draft burners. To enable the standard blower and furnace casting to be used and still to allow the furnaces to be entered, a hinged furnace access ring is used. This ring is shown in Fig. 16. It is arranged to carry on the inside a fire tile moulded in place from plastic refractory material. The furnace casting and turbo-blower are secured to the outside with hinges. By swinging the furnace access ring away from the furnace front, the furnace may be entered without difficulty.

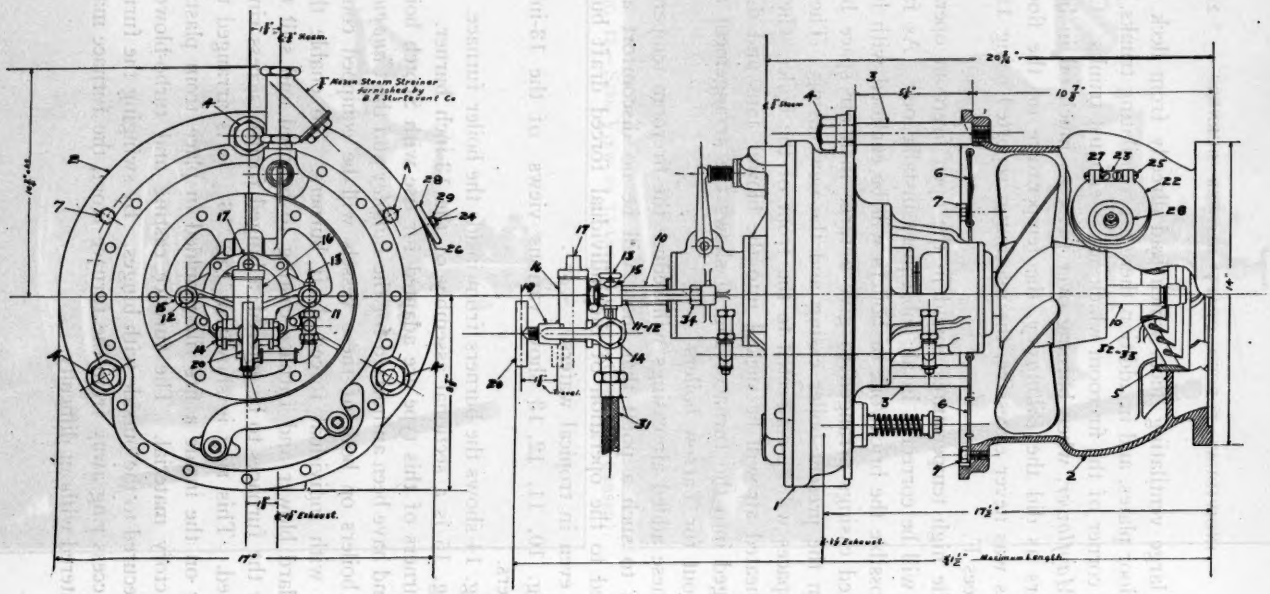


FIG. 15.—SECTIONAL ASSEMBLY OF 13-INCH BURNER.

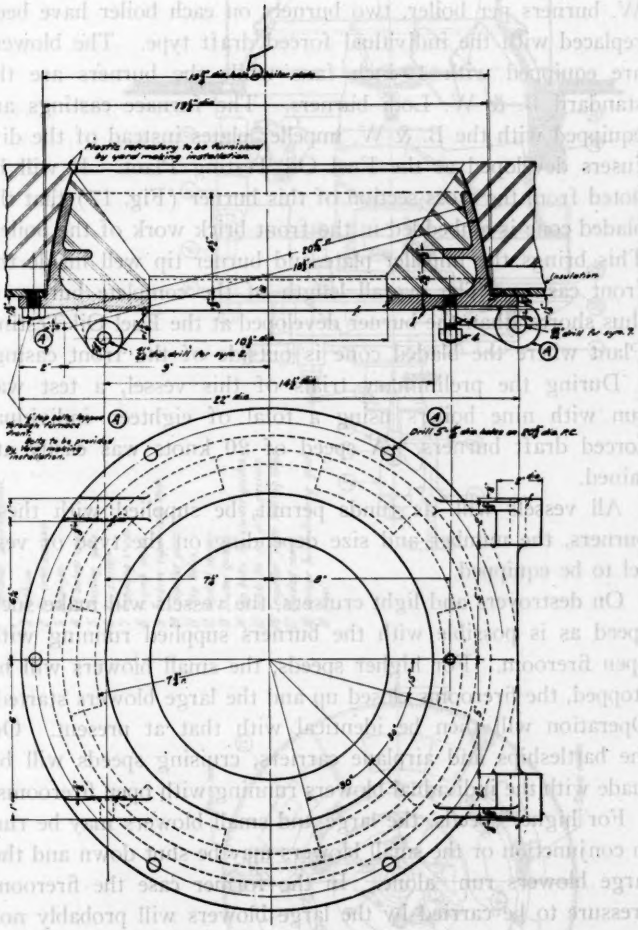


FIG. 16.—FURNACE ACCESS RING FOR SCOTCH BOILERS.

On the *Raleigh,* which is equipped with six Lodi size B. & W. burners per boiler, two burners on each boiler have been replaced with the individual forced draft type. The blowers are equipped with 19-inch fans while the burners are the standard B. & W. Lodi burners. The furnace castings are equipped with the B. & W. impeller plates instead of the diffusers developed at the Fuel Oil Testing Plant. It will be noted from the cross-section of this burner (Fig. 17) that the bladed cone is embedded in the front brick work of the boiler. This brings the impeller plate and burner tip well inside the front casing. The overall length of the complete burner is thus shorter than the burner developed at the Fuel Oil Testing Plant where the bladed cone is outside of the front casing.

During the preliminary trials of this vessel, a test was run with nine boilers using a total of eighteen individual forced draft burners. A speed of 20 knots was easily attained.

All vessels will, as funds permit, be supplied with these burners, the number and size depending on the type of vessel to be equipped.

On destroyers and light cruisers, the vessels will make such speed as is possible with the burners supplied running with open fireroom. For higher speeds, the small blowers will be stopped, the firerooms closed up and the large blowers started. Operation will then be identical with that at present. On the battleships and airplane carriers, cruising speeds will be made with the individual blowers running with open firerooms.

For higher speeds, the large and small blowers may be run in conjunction or the small blowers may be shut down and the large blowers run alone. In the former case the fireroom pressure to be carried by the large blowers will probably not exceed two inches of water, as the pressure needed is only that necessary to supply air for combustion and that necessary to maintain an excess of fireroom pressure over that built up in the furnace.

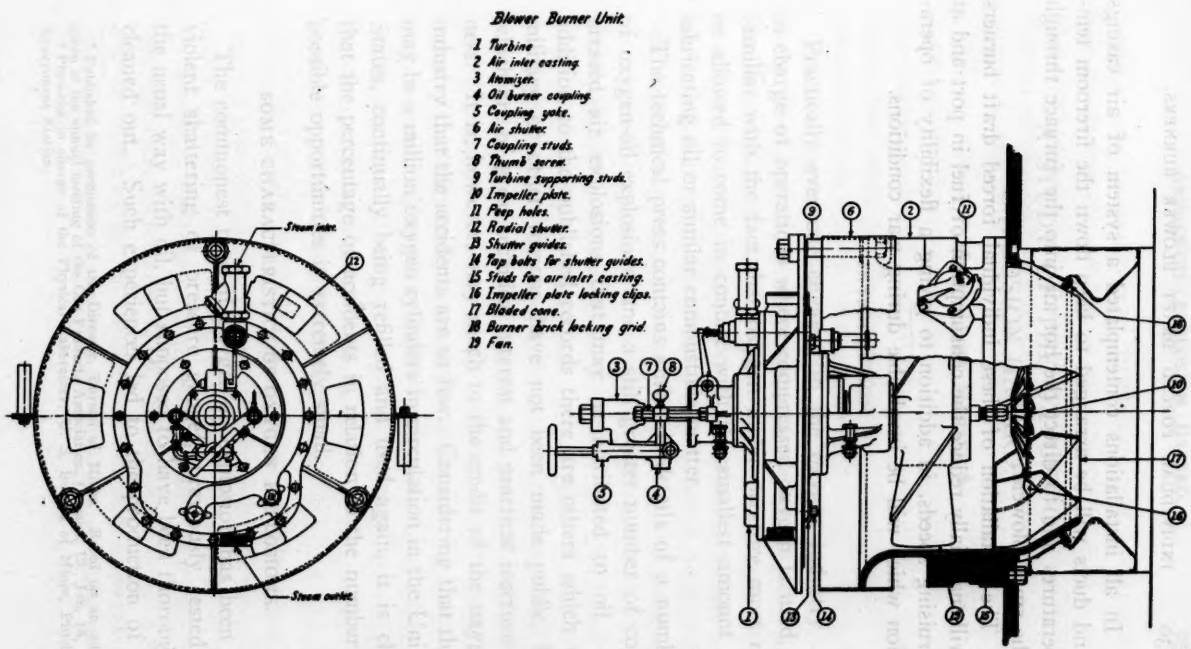


FIG. 17.—CROSS-SECTION OF 19-INCH BURNER B. & W. DESIGN (STURTEVANT BLOWER).

In all installations contemplated a system of air casings and ducts will be arranged to keep down the fireroom temperatures and to direct the hot air into the furnace through the small blowers.

The installation of these individual forced draft burners will materially reduce the consumption of fuel in port and at cruising speeds, in addition to giving a flexibility of operation which will be invaluable during war conditions.

A STUDY OF THE OXYGEN-OIL EXPLOSION HAZARD.*

By Mayo Dyer Hersey.†

INTRODUCTION.

Practically everyone engaged in the oxygen industry, or in charge of operations where compressed oxygen is used, is familiar with the fact that oxygen under pressure must not be allowed to come in contact with the smallest amount of lubricating oil or similar combustible matter.

The technical press contains authentic details of a number of oxygen-oil explosions, and a still greater number of compressed air explosions that may be attributed to oil. In addition to the published records there are others which for military or other reasons have not been made public, but which are of great technical interest and practical instructiveness. It is, however, very much to the credit of the oxygen industry that the accidents are so few. Considering that there may be a million oxygen cylinders in circulation in the United States, continually being refilled and used again, it is clear that the percentage of accidents in relation to the number of possible opportunities is extremely small.

SOME CHARACTERISTICS OF OXYGEN EXPLOSIONS.

The commonest type of oxygen-oil explosion has been the violent shattering of pressure gauges, previously tested in the usual way with oil, but supposed to have been thoroughly cleaned out. Such experiences led to the production of the

* Published by permission of the Director, Bureau of Mines. Based on an address given at the annual meeting of the Gas Products Association, Chicago, Ill., Jan. 18, 1924.

† Physicist in charge of the Physical Laboratory, U. S. Bureau of Mines, Pittsburgh Experiment Station.

solid-front oxygen gauge with its engraved warning, "Use no oil." This particular type of accident is now much more rare, but in general the majority of accidents do occur immediately following the sudden opening of valves, or under other circumstances permitting local development of heat by friction or compression.

In addition to oxygen-oil explosions it is important to guard against explosions due to the mixture of oxygen with hydrogen, acetylene, or other combustible gases, and against the simple mechanical failure of cylinders or other containers due to excess pressure or imperfect construction.

A peculiar characteristic of oxygen explosions is the astonishing amount of mechanical power developed (that is, rate of liberation of energy) in relation to the small amount of combustible matter probably present. The violence of the detonation and general destructive effect seem out of proportion to the actual cause. If the containing vessel is made of steel it may burn like celluloid, or be splintered into brittle fragments like glass.

The term "oxygen explosion" is, of course, simply an abbreviated expression. Oxygen alone is as harmless as nitrogen and can explode only by going into chemical combination with some other substance, such as oil, or oil vapors, hydrogen gas, certain packing materials, rubber or fibre gaskets, or possibly iron and steel.

OCCASION FOR THIS INVESTIGATION.

Considerable attention was drawn to the subject of oxygen explosions in consequence of the disaster at Harvard University in May, 1922, when two persons were instantly killed, others injured, and extensive property damage done. In the course of the inquiry Walter M. Wedger, chemist for the State of Massachusetts, clearly pointed out the fact that no one knows what are the technical limiting conditions for explosibility of oil in oxygen.

This is a fair question to ask, for we do know the limiting conditions for explosibility of hydrogen in oxygen, and that knowledge has been of distinct value. Can similar knowledge be secured for the oxygen-oil combination?

On the initiative of Dr. Charles E. Munroe, Chairman of the Committee on Explosives Investigations, National Research Council, the problem of oxygen-oil explosions was immediately taken up, and referred to the Pittsburgh Experiment Station of the Bureau of Mines. Work was actively begun somewhat more than a year ago, with the financial support of the Navy Department and others interested.

It was agreed that our first effort should be to produce actual explosions in the laboratory, by trying different conditions that might be thought favorable—that is, favorable for the spontaneous* detonation, or, at any rate, spontaneous combustion of oil in oxygen. However, at a conference of those officially interested (November 21, 1922) it was decided not to go above an initial temperature of 60 degrees C. (140 degrees F.) for the oil sample (unless this should result from its own oxidation), and therefore to exclude the familiar case of sudden compression. This was adopted, in fact, as our standard working temperature for it was believed to represent the maximum temperature that could be reached by an oxygen cylinder standing undisturbed in the sunshine under ordinary circumstances. We of course expected later to vary the temperature, both by sudden compression and by external heating, but were asked to defer this until explosions had first been secured at the lower temperature.

PRELIMINARY EXPERIMENTS.

It seemed simple enough to get an explosion in this way by sufficiently increasing the pressure, and hence the concentration of the oxygen, while keeping the temperature constant. But unfortunately there are hazards in research work no less

* The term "Spontaneous" will be used simply to denote ignition without the help of spark or flame.

than in industrial practice—referring now, not to the hazard of accidents from apparatus imperfectly constructed, though of course that has to be reckoned with and avoided in all oxygen experiments: but simply to the hazard of negative results. This disappointment is not at all unfamiliar in research work.

For a long period our oil samples refused to explode, although the pressures were carried well beyond 2,500 pounds per square inch. Naturally an air of mystery and excitement surrounded these first attempts. After turning on the long handled valves and pulling wires from behind a barricade we listened anxiously for results, while visitors gathering near placed small bets on the probable outcome. In these tests not only mineral lubricating oil, but also kerosene and linseed oil were tried, and the samples were well spread out over paper or cotton waste, both separately and in the presence of iron rust, copper acetylide, and water vapor; and the container volume was varied from less than 1 up to 1,000 cubic centimeters. In no case after opening the bomb did the oil show any evidence of combustion or did the gas analysis indicate carbon dioxide.

These facts satisfied us fully on the low temperature work. We proceeded to elevate the temperature, and in less than two hours after gently starting the heater, the first explosion occurred with a sharp report. This was with linseed oil at a temperature of 120 degrees C. (about 250 degrees F.).

There was no mistaking the fact that we now had a positive result, and at no time since has there been any failure to produce an explosion when desired. Of course, the violence of the detonation is greatly influenced by the amount of initial oxygen pressure.

To register the final explosive pressure developed a special gauge was designed, known as our multiple-disc pressure gauge. This little instrument showed that an initial charging pressure of the amount usually carried in commercial cylinders, 1,800 to 2,000 pounds per square inch, may lead to an explosion

pressure of 20,000 pounds per square inch when a few drops of mineral lubricating oil are sufficiently warmed up to explode spontaneously in our standard steel bomb of 4 cubic centimeters volume.

The question now arises, what possible chance can there be in actual practice of approaching anywhere near this danger-temperature of 250 degrees F.? The answer is that there are plenty of chances to exceed this temperature, as every compressor manufacturer knows, for it is not the temperature of the surrounding atmosphere, or even of the metal parts, that we are concerned with, but the temperature of the oil film itself. Oil is a poor conductor of heat. That side of the film which is in contact with the oxygen might come up to its ignition temperature while the other surface is still in contact with cool metal. There are two principal ways in which oil, if present, can quickly be brought to the danger temperature: (1) by slow leakage of high pressure gas carrying liquid lubricant with it through some very minute opening—that is, by internal friction, which always generates heat; and (2) by sudden compression of the layer of oxygen nearest to the end of a closed passage, whether due to the impact of high velocity oxygen rushing in from a quickly opened valve, or to direct connection with the compressor, as in the case of the Harvard explosion.

In carrying out the Bureau of Mines experiments thus far described, under the direction of A. C. Fieldner, Superintendent of the Pittsburgh Experiment Station, I was assisted by J. J. Jakosky, S. H. Brooks, and E. W. Butzler of the Physical Laboratory staff. Immediately following the first experiments that led to positive results, it was decided to make a division of labor. While Jakosky and Butzler branched out on a new line and determined the spontaneous ignition temperature of metals in oxygen at different pressures, Brooks continued the observations on explosibility of oil with the original 4 cubic centimeter bomb and multiple disc gauge, and special experiments on the flow of gases were undertaken by

Jakosky and the writer, which are still in progress. The results so far obtained in these three directions may now be briefly described.

IGNITION OF METALS.

As shown in Figure 1, the temperature for spontaneous ignition of iron drops from 1,700 degrees to 1,100 degrees F. when the oxygen pressure is increased from atmospheric pressure to 2,000 pounds per square inch. Brass and copper are safe at higher temperatures than iron, and for pressures above 1,200 pounds per square inch their ignition temperatures fall off very little indeed with further rise of pressure. These experiments were made on metal in the form of wire, heated very gradually by passing an electric current through, and raising ampèrage until suddenly the wire ignites and automatically breaks the circuit. The oxygen pressure was meas-

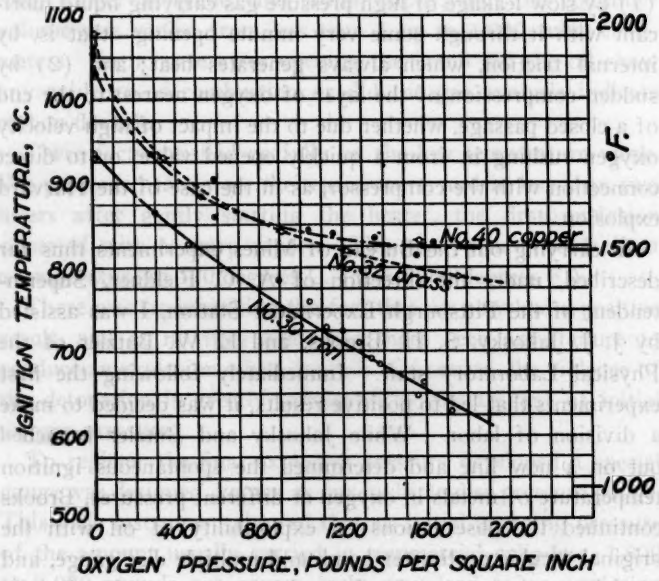


FIG. 1.—IGNITION OF METALS.

ured with a gauge after being admitted to a large metal bomb containing the wire specimen. The wire temperature was measured electrically by resistance observations—that is, from the voltmeter and ammeter readings at the instant preceding ignition, after previously calibrating the wire in a high-temperature furnace to determine its resistance at different temperatures. Samples of lead melted off at a constant temperature of about 630 degrees F. at all pressures. These results suggest that brass and copper are probably much safer materials to use for manifold construction and other apparatus than iron or steel; because the observed ignition temperature of the iron is far below the theoretical temperature (1,700 degrees F.) that could be reached by instantaneous compression to 2,000 pounds per square inch, while that of copper is but slightly below. No oil was present in these experiments.

CRITICAL TEMPERATURE FOR EXPLOSIBILITY OF OIL.

In continuing the oil experiments four typical samples were tested—two mineral oils (Mobiloil A and kerosene), one animal oil (sperm), and one vegetable oil (linseed). For comparative results these samples were all tested in the same 4 cubic centimeter bomb, the same weight of oil being taken each time, as nearly as practicable, and the oil being well spread out on pure asbestos. No catalyzer of any kind was intentionally present. From the numerous observations taken it was established that the spontaneous ignition temperature of oil is a constant—that is, independent of the oxygen pressure over the full range tested (from atmospheric pressure up to 3,600 pounds per square inch). In this respect the ignition of oils stands in marked contrast to the ignition of metals. The apparatus is shown in Figure 2. Oxygen pressure was measured on a gauge before heating the bomb, which, while being heated, was closed off from the oxygen supply line by means of a suitable valve. The actual pressure in the bomb just preceding explosion was computed from the initial gauge

pressure by making proper allowance for the rise of temperature of the confined volume of oxygen in the bomb. The approximate oil temperature at the moment preceding ignition was observed by means of a thermocouple arranged to give the

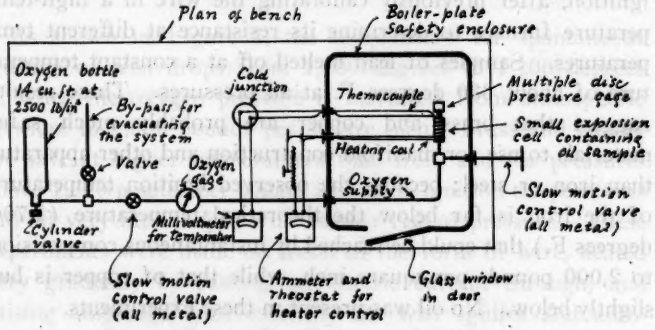


FIG. 2.—OXYGEN-OIL EXPLOSION APPARATUS.

temperature of the steel bomb, which was surrounded by the electrical heating coil employed for slowly raising the temperature of the bomb. The observed ignition temperatures are given in the following table to the nearest 5 degrees C. and 10 degrees F.

IGNITION TEMPERATURE

Oil	Degrees C.	Degrees F.
Kerosene...	175	350
Mobiloil A..	170	340
Sperm.....	135	280
Linseed.....	120	250

These figures refer to the very first explosion in each experiment. Successive explosions were produced in the same closed bomb by further raising the temperature, thus confirming reports of multiple explosions in connection with certain accidents.

The ignition temperature for a given oil may be different for different sized containers, or other conditions, such as are shown under the head of "Explosibility variables" in the outline of "Further Research Problems."

EXPERIMENTS ON THE FLOW OF GASES.

It has been noticed in practice that explosions and fires most often occur immediately after opening some valve, or during the operation of a compressor. For this reason experiments are being made to determine (1) the possibility of combustible matter being found in the oxygen jet passing out of a cylinder valve; (2) the amount of electrification that can be caused when oxygen flows through a valve or other fine opening; (3) the exact distribution of temperature in a stream of oxygen; (4) the possibility of ozone formation due to flow of oxygen, the ozone molecule (O_3) being much more active, chemically, than ordinary oxygen, and therefore liable to give a lower ignition temperature when in contact with oil. So far, these tests have indicated the following:

(1) A number of cylinders of atmospheric oxygen were selected at random, and the outflow passed through filter paper and into a U-tube immersed in liquid oxygen. A black metallic deposit gathered on the filter paper, while some oily or organic substance collected in very small amounts on the walls of the glass U-tube.

(2) Different gases passed through a pin-hole orifice under a pressure head of 1,800 pounds per square inch invariably showed a strong electrostatic charge when, and only when, either dust or liquid moisture passed through the orifice with the gas. This electrification may at times facilitate ignition.

(3) The temperature of a jet of oxygen escaping under a pressure of 2,000 pounds per square inch shows a "Joule-Thomson" temperature drop of 40 degrees C. (about 70 degrees F.) below the cylinder temperature after being brought to rest by impact. The stagnant film of gas which may be caught between this impinging jet and a stationary surface

may itself rise in temperature momentarily to a considerable and probably dangerous degree. The moving jet, at the point of its greatest velocity, experiences a further temperature drop in addition to the 70 degrees F. drop already mentioned. This particular temperature cannot be measured experimentally very well because, to do so, would require moving our thermometer with the velocity of the jet; but it can safely be calculated theoretically if the true velocity is known.

(4) Electrification and cooling of the jet both combine to favor ozone formation, and ozone has been thought to be present.

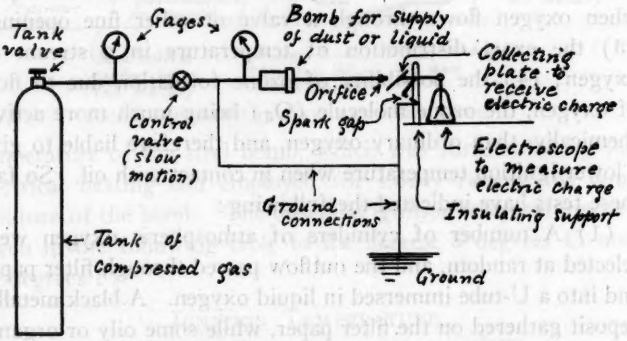


FIG. 3.—ELECTRIFICATION DUE TO FLOW OF GASES.

These last mentioned experiments, when completed and taken in conjunction with the ignition of metals and the ignition of oils, should provide a reliable basis of technical facts for reference in analyzing and explaining a good proportion of the accidents on record, and thus reduce the hazard in the future.

FURTHER RESEARCH PROBLEMS.

Considerable investigation relating to oxygen-oil explosions remains for the future, and will be of value not only in analyzing accidents, but also in developing safety precautions and

improving the efficiency of methods of handling and utilizing oxygen. These future problems are outlined to some extent below:

I. Ignition of metals.
   1. Extension to higher pressures.
   2. Effect of purity of oxygen, water-vapor, etc.
   3. Comparative safety of different grades of steel and non-ferrous alloys.
   4. Ignition of metal mass by propagation from splinters.
   5. Effect of oil on the metal surfaces.

II. Comparative explosibility of substances.
   1. Air compressor oils.
   2. Oxygen compressor lubricants.
   3. Valve packing, gaskets, etc.

III. Explosibility variables.
   1. Size, shape, and heat insulation of container.
   2. Catalyzers.
   3. Time periods, turbulence, etc.
   4. Amount and dispersion of oil.
   5. Effects of dust and of moisture.
   6. Nitrogen dilution.
   7. Effect of extreme high pressures.

IV. Thermodynamic problems.
   1. Temperatures due to sudden compression.
   2. Temperatures due to rapid flow and slow leakage.
   3. Ignition by sparks.
   4. Solubility and vapor pressure.
   5. Explosive energy available.

V. Elasticity problems.
   1. Energy required to fracture metal.
   2. Measurement of explosive pressures.

PRACTICAL SAFETY PRECAUTIONS.

A provisional schedule of safety precautions for use on board ship, or at Navy Yards and plants where oxygen is employed, may be based on information already available without waiting for the final solution of the foregoing problems. The following are taken from a list of precautionary suggestions approved September 30, 1922, by the Superintendent, Pittsburgh Experiment Station, U. S. Bureau of Mines, together with rules for the safe handling of oxygen cylinders compiled by the Gas Products Association, and Navy Department Instructions:

(1) Use only those cylinders which conform to the Interstate Commerce Commission "Regulations for the Transportation of Explosives and Other Dangerous Articles." In accordance with Navy Department practice, if a cylinder is discovered not having markings thereon as required,* it is immediately surveyed for destruction.

(2) Allow no opportunity for different gases to mix. Storage of oxygen cylinders in the same place with cylinders of combustible gases should not be permitted. Analyze the first gas taken out of new cylinders before using if there is ground for doubt as to its freedom from impurities; this applies particularly to oxygen.

(3) Handle cylinders with extreme care so as to avoid jarring or dropping or any mechanical injury to the cylinder body, valve, or safety plug.

(4) Oxygen cylinders should not be exposed to excessive heat. Air compressors must be properly cooled. Oxygen apparatus of all kinds should be so adjusted and used as to reduce the possibilities for local development of heat internally, whether by leakage, mechanical friction, sudden compression, or otherwise.

* See paragraphs 556 and 557, (Revised Edition, effective Jan. 1, 1923; available from Government Printing Office, Washington, D. C.). These regulations are based on the recommendations of the Bureau of Explosives and contain full specifications for the manufacture and acceptance of cylinders, periodical annealing, hydrostatic tests, etc.

(5) Contact of high pressure oxygen with combustible substances, including fiber, leather, and rubber may at times be dangerous. Oxygen fittings must not be contaminated by oil; this precludes the use of oil in machining oxygen parts, and also requires, as a rule, that no other compressed gas be admitted to any oxygen apparatus at any time, since oil is used as a lubricant in compressing the other gases and might thus be carried over into the oxygen line.

In short, it is necessary for the safe use of oxygen to avoid (1) imperfect cylinders, (2) gas mixtures, (3) mechanical injury, (4) heat, and (5) combustible matter, especially oil.

CONCLUSION.

Further experiments of the type reported in this paper will be helpful in the detailed formulation of safety codes and in developing protective devices for reducing the hazard from compressed oxygen and compressed air explosions. It is also evident that the oxygen-oil explosion problem has very interesting relationships to several other branches of scientific research, such as the development of military and mining explosives, and the theory of the Diesel engine and other internal combustion motors. The physical investigation of oxygen-oil explosions, in fact, appears to be a field of sufficient practical importance to warrant considerable further study.

STRESS DISTRIBUTION IN CRANKSHAFTS AND ITS RELATION TO CRANKSHAFT FAILURES.

By D. J. McAdam, Jr., and G. F. Wohlgemuth.*

HISTORY OF INVESTIGATION.

During a period of about ten years a number of failed crankshafts have been received for investigation at the Naval Engineering Experiment Station. Recently a large proportion of these have been aeroplane crankshafts.

In investigating such cases of failed material there are three questions to be considered:—(*a*) Does the quality of the metal conform to the specifications? (*b*) Is the design of the piece, as regards dimensions and specified quality of material, suitable for the intended service? (*c*) Has the piece been subjected to abuse or abnormal service?

Examination of these failed crankshafts has shown that in only a few cases is the material defective or inferior to the specified quality. In the great majority of such shafts the cause of failure must be assigned to some deficiency in design. Such deficiency in turn can usually be traced to lack of data in regard to the amount and distribution of the stresses in rapidly rotating objects of such complicated form. In crankshafts under service conditions there is no doubt that frequently the moment of torque and consequently the stress greatly exceed any value obtained by the usual methods of calculation. The distribution of stress in a crankshaft, moreover, is evidently very different from the usual assumptions of the designing engineer.

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In one of the Experiment Station's reports over a year ago aeroplane crankshaft failures were discussed and attention was called to the frequency with which fractures occurred in a web at its junction with either pin or journal. It was pointed out that the fractures usually start at the re-entrant angle between a web and either a journal or a pin and extend diagonally through the web into the axial oil hole in pin or journal. The stress concentration at the re-entrant angle and the slight resistance of this section to longitudinal bending of the shaft were also discussed. It was suggested that failures might be accounted for by longitudinal bending of the shaft, which would be possible especially after bearings had been worn down.

On the other hand, it was noted that many fractures of submarine crankshafts have occurred through journals. Such failures often start at regions of stress concentration such as oil holes.

Recently it has become more evident that there is great need for an analysis of the stress distribution in crankshafts. Some designing engineers argue that the only important stresses, due to torque in a crankshaft as a whole, are torsional stresses in journals and pins and transverse bending stresses in the webs. They assert that the frequent failures through the webs of aeroplane crankshafts are due to transverse or edgewise bending stresses in the webs. Other designing engineers have also asserted that there could be no important torsional stresses in the webs of a crankshaft due to torque in the shaft as a whole.

DISCUSSION OF CAUSES OF CRANKSHAFT FAILURES.

The evidence obtained by a study of the surfaces of fracture of aeroplane crankshafts does not support the theory of those engineers who assert that the failures are usually due to edgewise bending stresses in the webs. Fatigue fractures due to edgewise bending stresses in a web would start at the edges of the web and progress towards the axis. The last metal to yield would be that at the re-entrant angle between the web

and the pin or journal. Examination of the surface of fracture, however, shows that with a few possible exceptions found in crankshafts of old design the fractures start at the re-entrant angle and extend through the web diagonally into the axial oil hole in the pin or journal. The edges of the web are usually about the last portions to fail.

It has occasionally been possible, moreover, to find a shaft in which a fracture has started at the re-entrant angle but has extended only partially through the web. In Fig. 1 is shown a Curtiss shaft of old design in which such a crack exists. This shaft had fractured completely at the junction of a web and pin. By application of a mixture of iron filings and kerosene to the magnetized shaft the crack shown in Fig. 1 was found at the junction of another web and pin. Such a fracture could not have been caused by transverse bending stresses in the web.

In Fig. 2 is shown another such partial fracture of a failed Liberty crankshaft which was not received at the Experiment Station. Examination of this photograph shows clearly that the widest part of the crack (the point at which the fracture started), is so near the line joining the axes of journal and pin that transverse bending stresses could not have been an important factor in starting the crack. This photograph adds to the weight of evidence that crank web failures almost invariably start at the re-entrant angle between web and pin.

A fracture starting at the re-entrant angle between a web and either pin or journal could be due to either of two kinds of stress. It could be due to bending of the web longitudinally with reference to the shaft or it could be due to torsional stresses in the web. While some of these failures are no doubt due to longitudinal bending, it seems advisable to consider carefully the possibility that torque applied to a crankshaft as a whole may produce relatively high torsional stresses in the webs. Such stresses would become of great importance in a shaft rotating at a critical speed.

FIG. 1.

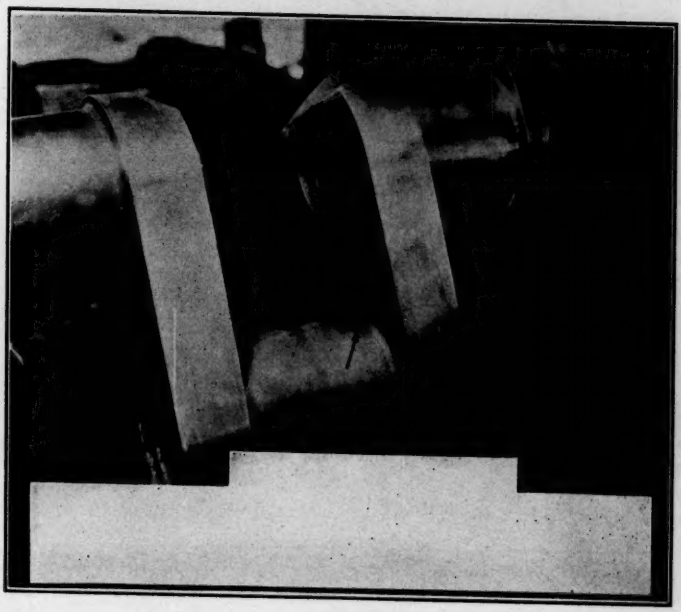


FIG. 2.

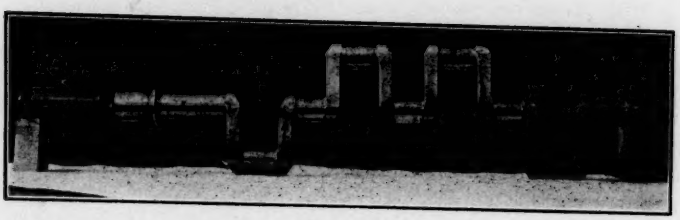


FIG. 10.

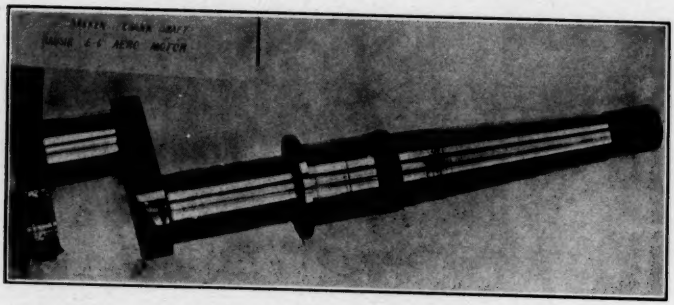


FIG. 11.

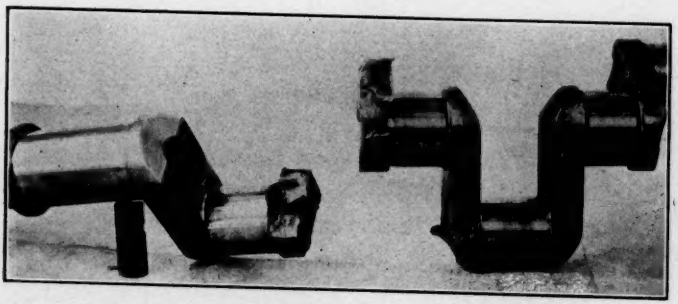


FIG. 12.

STRESS DISTRIBUTION IN CRANKSHAFTS AS AFFECTED BY RESTRAINT OF JOURNALS.

In the following discussion of stress distribution it will be assumed for the sake of simplicity that the shaft is subjected to torque of equal value applied to the end journals and that the stresses throughout the shaft are due solely to this torque. The effect of local variations in stress due to thrust of explosive forces, action of centrifugal forces, etc., will be neglected.

The following statement was recently noted:—"If a member composed of elements parallel to, and others normal to, an axis, be subject to pure torsion about that axis, then the stresses and deflections in the longitudinal elements will be purely torsional and the stresses and deflections in the lateral elements will be purely bending." Further, it has been asserted that these conditions apply to the Liberty crankshaft whether or not it is restrained by its seven bearings. There is no doubt that a crankshaft subjected to free torsion and with lateral displacement of journals unrestrained would behave in accordance with the above quoted statement. It will now be shown, however, that the behavior of a crankshaft held in its bearings may be different from that of a shaft not thus restrained.

If in the crankshaft section shown in Fig. 3, lateral displacement of the two journals is not prevented, a clockwise torque in the journals would be accompanied by an equal clockwise torque in the pin and by a pure bending moment in the webs. By this action the two journals would be displaced so that they would no longer be exactly in line. If, however, the journals be held so rigidly in their bearings that lateral displacement or lateral deflection of any kind in the journals is prevented, it can be shown that clockwise torque in the journals produces counter clockwise torque in the webs and vice versa.

In calculating the quantitative stress relationship in pin, webs and journals, the stiffness of these parts must be taken into account. If stiffness relationship could be neglected the

torque and shear relationship could be expressed very simply in the terms illustrated in Fig. 3. As illustrated in Fig. 3, the torque in the journals could then be expressed in terms of a force F, acting in opposite directions at the junction of the pin with each of the two webs, multiplied by its arm R:—

$$M \text{ (torque in the journal)} = FR$$

The torque in the webs could be calculated from the fact that the couple F*l* acting at the two extremities of the pin is balanced by the torque in the two webs. Therefore, if the torque in a web be represented by T, F*l*=2T or T=F*l*/2. Since M=FR, T=M*l*/2R.

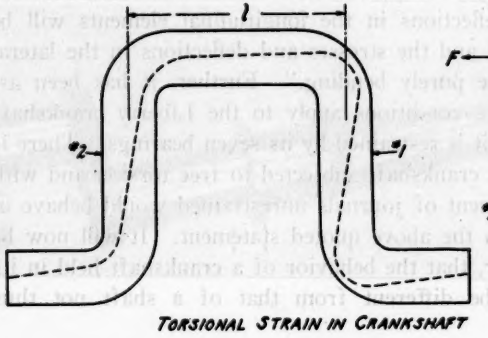
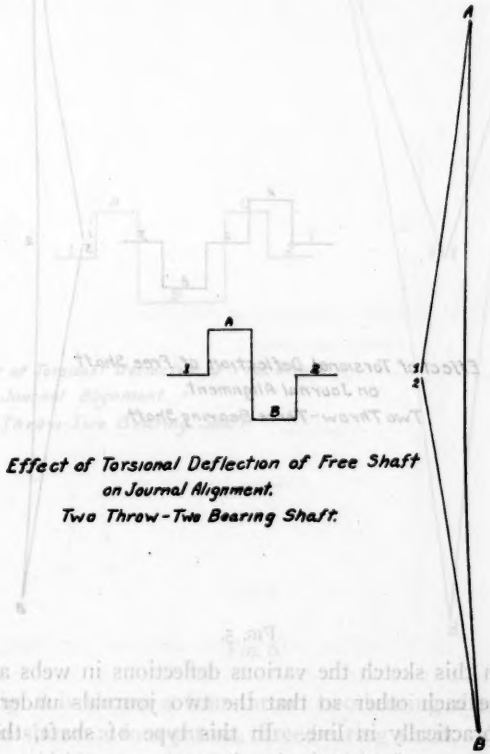


FIG. 3.

Under these conditions there would be no torque in the pin. There would, however, be shearing stress in pin and webs equal to the force F divided by the sectional area. Actually the effect of the stiffness relationship is usually such that in a shaft restrained as described above, the web torque would be somewhat less than M*l*/2R and there would be some torque in the pin. The effect of deflections in pin and webs will be discussed later.

It has been asserted by some engineers that lateral restraint of the journals of a crankshaft does not occur to a sufficient degree to be of importance. In support of this assertion photo-

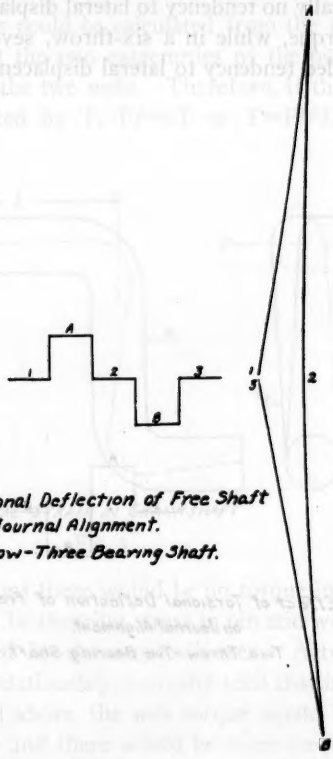
graphs have been presented which show that in a model of a four-throw, three-bearing shaft under torque the pins and journals remain apparently parallel. From these premises it is argued that there is no important torque in the webs of a Liberty crankshaft. In answer to this argument, however, it will be shown that in the four-throw, three-bearing shaft there is practically no tendency to lateral displacement of the journals under torque, while in a six-throw, seven-bearing shaft there is a decided tendency to lateral displacement of journals under torque.



Effect of Torsional Deflection of Free Shaft on Journal Alignment. Two Throw-Two Bearing Shaft.

FIG. 4.

In Figs. 3 to 9 inclusive are shown sketches illustrating the lateral displacements of journals in free shafts of various types under clockwise torque. In Fig. 4 is shown a two-throw, two-bearing shaft and on the right is shown on an exaggerated scale the effect of bending of webs and twist of pins on the alignment of the two unrestrained journals. As

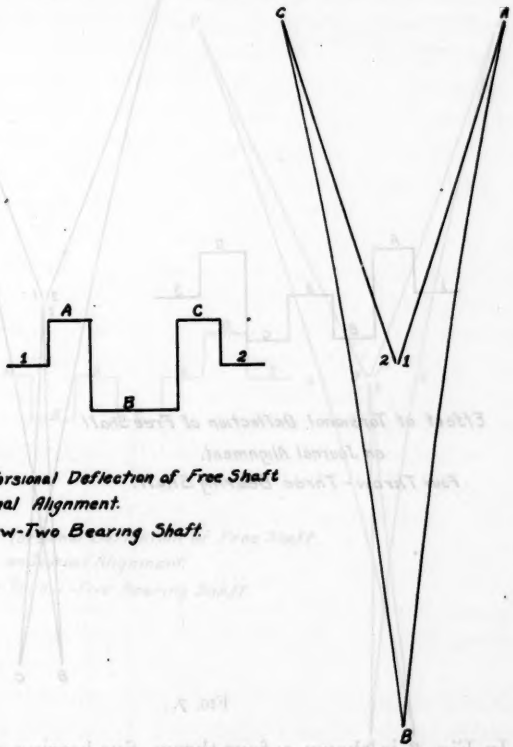


Effect of Torsional Deflection of Free Shaft on Journal Alignment. Two Throw-Three Bearing Shaft.

FIG. 5.

shown in this sketch the various deflections in webs and pins neutralize each other so that the two journals under torque remain practically in line. In this type of shaft, therefore, even when restrained by its bearings, there could be no important torque in the webs due to torque in the journals.

In Fig. 5 is shown a two-throw, three-bearing shaft. This differs from the shaft shown in Fig. 4 only in the fact that the former has another bearing in the center. As shown in the sketch, the middle bearing is deflected out of line with the two end bearings. In this type of shaft, therefore, restraint of lateral displacement of journals would cause torque in all the webs.

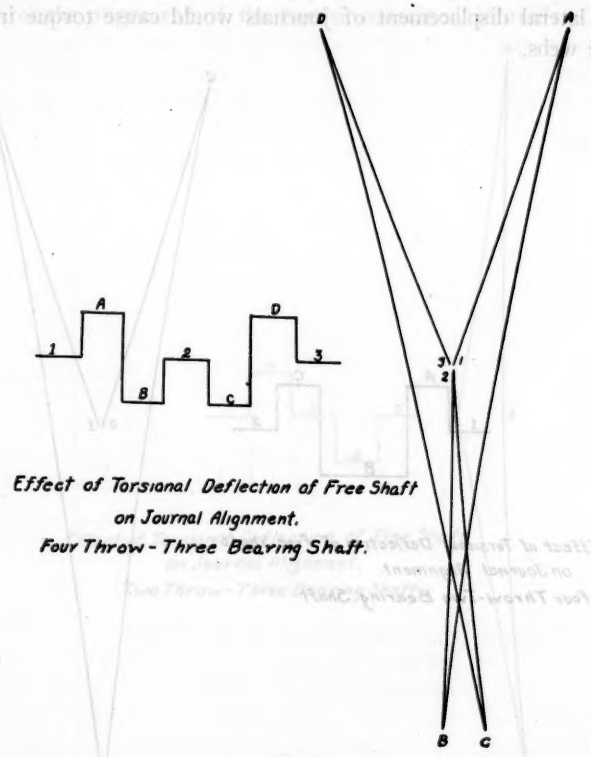


Effect of Torsional Deflection of Free Shaft on Journal Alignment. Four Throw-Two Bearing Shaft.

FIG. 6.

In Fig. 6 is shown a four-throw, two-bearing shaft and in Fig. 7 is shown a four-throw, three-bearing shaft. As shown in the enlarged sketches at the right of these figures, the

deflections of pins and webs in these two types of shaft neutralize each other. In these shafts under torque therefore, the journals even when under no restraint, remain in line. In these types, therefore, there can be no important torque in the webs.



Effect of Torsional Deflection of Free Shaft on Journal Alignment. Four Throw - Three Bearing Shaft.

FIG. 7.

In Fig. 8 is shown a four-throw, five-bearing shaft. This shaft differs from the one shown in Fig. 7 in that it has two additional journals shown at 2 and 4 in Fig. 8. As shown in the sketch on the right, the deflections in pins and webs are such that in the free shaft under torque journals 2 and 4 are

thrown out of line with the other three journals. Restraint of this lateral displacement would produce torque in all the webs of this type of shaft. It is worthy of note that the failed shafts shown in Figs. 2, 10 to 14 are of this type and that these failures occurred in webs. There are a number of similar examples at the Experiment Station, but space does not permit the reproduction of photographs of them.

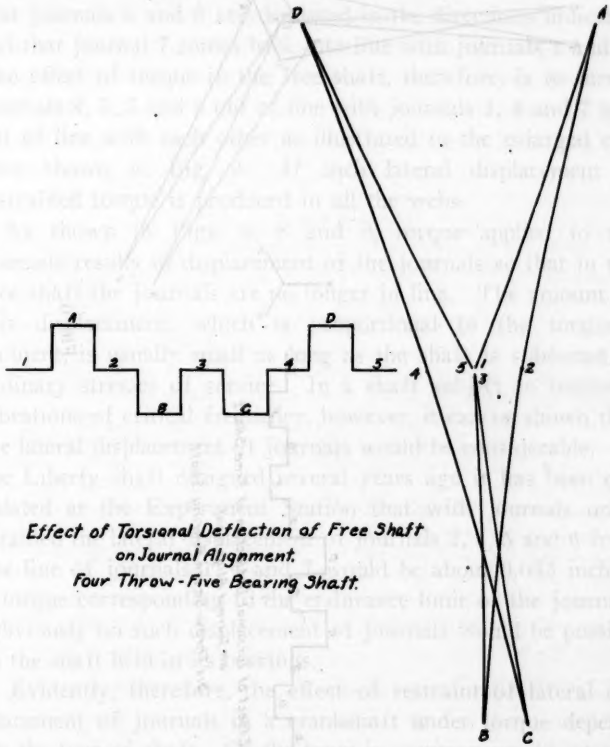
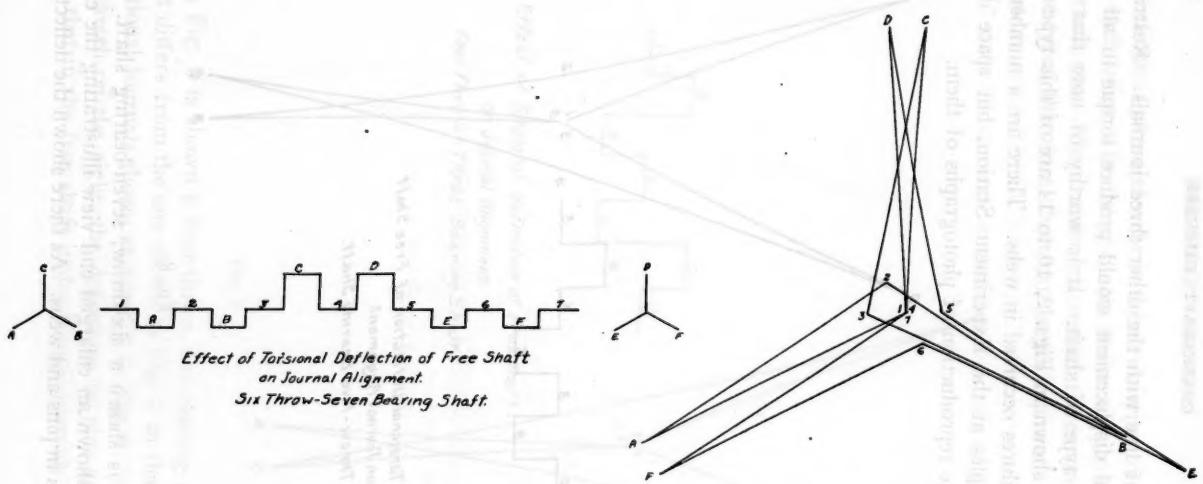


FIG. 8.

In Fig. 9 is shown a six-throw, seven-bearing shaft. On the right is shown an enlarged end view illustrating the effect of deflections in pins and webs. As there shown the deflections



Effect of Torsional Deflection of Free Shaft on Journal Alignment. Six Throw-Seven Bearing Shaft.

FIG. 9.

in the pin A and its two adjacent webs displaces journal 2 laterally with reference to journal 1. Similarly the deflections in pin B and its adjacent webs displaces journal 3 laterally with reference to journal 2 and in the direction shown. Similarly deflection of pin C and its adjacent webs displaces journal 4 with respect to journal 3 in such a direction that journal 4 is now in line with journal 1. In the same way it can be shown that journals 5 and 6 are displaced in the directions indicated and that journal 7 comes back into line with journals 1 and 4. The effect of torque in the free shaft, therefore, is to throw journals 2, 3, 5 and 6 out of line with journals 1, 4 and 7 and out of line with each other as illustrated in the enlarged end view shown in Fig. 9. If such lateral displacement is restrained torque is produced in all the webs.

As shown in Figs. 5, 8 and 9, torque applied to the journals results in displacement of the journals so that in the free shaft the journals are no longer in line. The amount of this displacement, which is proportional to the torsional moment, is usually small as long as the shaft is subjected to ordinary stresses of service. In a shaft subject to torsional vibrations of critical frequency, however, it can be shown that the lateral displacement of journals would be considerable. In the Liberty shaft designed several years ago it has been calculated at the Experiment Station that with journals unrestrained the lateral displacement of journals 2, 3, 5 and 6 from the line of journals 1, 4 and 7 would be about 0.035 inch at a torque corresponding to the endurance limit of the journals. Obviously no such displacement of journals would be possible in the shaft held in its bearings.

Evidently, therefore, the effect of restraint of lateral displacement of journals in a crankshaft under torque depends on the type of shaft. Of the types in common use in internal combustion engines, the six-throw, seven-bearing shaft and the four-throw, five-bearing shaft are evidently subject to torque in the webs while the four-throw, three-bearing shaft under the same conditions is not subject to web torque. To de-

termine the possible stresses in the webs under those conditions the stress-strain relationships in the various crankshaft parts must be considered.

STRESS-STRAIN RELATIONSHIP IN TYPICAL NAVAL CRANKSHAFTS.

To determine quantitatively the possible effect of lateral restraint of journals on the stress distribution in various typical crankshafts, calculations have been made using data obtained from various failed shafts received at the Experiment Station and from sketches of newer shafts. The chemical composition and physical properties of the various shafts to be considered are given in Table I. The first half of Table I gives data in

TABLE I
Crankshaft Material
Chemical Composition and Tension Test Results.

Crankshaft and year received.	Chemical Composition.							Tension Test Results.			
	C	Mn	P	S	Ni	Cr	Va	Ultimate strength	Yield Point	Elongation per cent	Reduction of area per cent
Hall Scott*, 1917	.32	.62	.012	.044	1.49	.51		125,700	97,900	16.5	56.2
								108,900-	78,440-	13.0-	42.6-
Reusie*, 1921	.45	.80	.018	.04		1.10	.16	121,000	98,500	16.5	37.6
Liberty*, 1918	.39	.60	.011	.027	1.68	.46		137,800	115,000	19.5	58.1
Liberty* 1922	.29	.64	.042	.06	1.13	.79		150,000	125,000	17.	54.
Liberty** 1925											
Curtiss 1917	.35-	.60-	.04	.04	1.00-	.75-		135,000-	115,000-	17.	55
	.45	.80	(Max.)	(Max.)	1.50	1.00		145,000	130,000	20	60
Curtiss, 1922	.35-	.60-	.04	.04	1.00-	.90-					
	.45	.80	(Max.)	(Max.)	1.50	1.25		130,000	110,000	16	54
Curtiss.** 1922											
Lawrence, 1922-23	.35-	.50-	.040	.045	1.25-	.65-					
	.45	.80	(Max.)	(Max.)	1.75	.75		140,000	120,000	19	57
Van Blerck*,1922	.43	.63	.017	.021	1.41	.65		99,000	65,500	22.5	56.0
Sub.S-50, 1922	.40	.64	024	.031			.19	75,000	40,000	25	40
Sub.O-1*, 1923	.347	.771	.049	.028	3.05			94,750	47,700	23.0	36.2
USS PADUCAH*,1915	.546	.32	.015	.016	0.19			69,300	56,300	31.5	63.8
USS C-4, 1919	.41	.32	.007	.022	3.27			85,000	50,000	26.0	55.0
Elec.Boat Co., S Type.***. 1921								70,000	35,000	25	
Subs V-2 & V-3 1922	.30-	.50-	.04	.04	.50	.95-	.14-				
	.45	.80	(Max.)	(Max.)	(Max.)	1.15	.20	100,000	80,000	20	50
(See Note 1) Sub K-6, 1915	.257	.64	.029	.036	2.72			90,000	70,000	20	
(See Note 2) Sub K-7, 1916								98,500	65,500	23.5	39.9

* Engineering Experiment Station Results. Other values as specified or Manufacturer's Results.

** Specifications not received. No material available for test.

*** Specifications state "carbon steel, annealed". No reduction of area specified.

Note 1. Sub. K-6 chemical analysis made at Engineering Experiment Station, physical properties as specified.

Note 2. Material specifications are the same as those for Sub. K-6.

regard to aeroplane crankshafts and the last half of this table gives data in regard to crankshafts of various Naval vessels.

As shown in Table I, there are wide differences in physical properties of these shafts, all of which are subjected in service to stresses of the same general nature and may be subjected to stresses of not greatly differing intensity. Though the great difference between the properties of aeroplane and submarine shafts is partly accounted for by the difference in the size of the two classes of forgings, much of the difference is due to the fact that in submarine shafts steel of a lower grade has been used.

TABLE II.

Section Moduli of crankshaft sections in lb. and inch units.

Crankshaft & year received	Torsion				Bending				Weakest Section
	Journal	Pin	Web		*Web (Longitudinal)		*Web (Transverse)		
			At Journal	At Pin	At Journal	At Pin	At Journal	At Pin	
Hall Scott, 1917	1.51	1.51	.53	.53	.33	.33	.79	.79	.23
Hausie, 1921	1.79	1.79	.59	.59	.35	.35	1.10	1.10	
Liberty, 1918	3.28	2.42	.77	.73	.47	.45	1.32	1.20	.32
Liberty, 1922	3.28	2.42	1.02	.98	.59	.57	2.12	1.97	
Liberty, 1923	3.36	2.55	1.04	1.04	.60	.60	2.19	2.19	
Curtiss, 1917	1.21	1.21	.28	.28	.16	.16	.84	.84	.16
Curtiss, 1922	3.62	2.09	.68	.68	.39	.39	1.66	1.66	
Curtiss, 1922	3.62	2.09	.76	.74	.43	.42	1.90	1.83	
Lawrence, 1922-23	1.73	1.41	**.67 L .38 S	**.67 L .38 S	**.40 L .22 S	**.40 L .22 S	**1.18 L .67 S	**1.18 L .67 S	.18
Van Blerck, 1922	2.23	2.23	.75	.75	.43	.43	1.66	1.66	
Sub.S-50, 1922	92.30	123.77	58.14	58.14	36.10	36.10	89.49	89.49	
Sub.G-1, 1913	23.89	21.01	13.07	13.07	3.19	8.19	19.36	19.36	
USS PADUCAH, 1913	24.50	28.22	16.57	16.57	11.39	11.39	20.25	20.25	
USS C-4, 1912 1921	14.15	13.66	9.51	9.51	5.99	5.99	13.78	13.78	
Electric Boat Co. 1922	98.37	98.37	48.20	48.20	29.30	29.30	80.70	80.70	
USS V-2 & V-3	153.5	168.5	84.10	84.10	43.22	43.22	138.24	138.24	

* The terms "Longitudinal" and "Transverse" refer to directions parallel and perpendicular respectively to the axis of the shaft.

** S = Small, L = Large.

As a preliminary to the discussion of stress-strain relationships in these shafts, the section moduli have been calculated and recorded in Table II. In addition to the moduli involved in torque of the shaft the moduli of bending of the webs in the direction of the shaft axis have been included. In Table

III are given the ratios of the various moduli to the torsional modulus of the journal. Table III has not been used in later calculations but is given here to illustrate the wide difference in these ratios for the different shafts. In general the ratios for the moduli of the web are considerably lower for the aero-

TABLE III.

Ratios of various moduli to torsional modulus of journal.

Crankshaft & year received.	Torsional Modulus of Pin.	Torsional modulus of Web.		Bending modulus of web. (Long.)		Bending modulus of web. (Trans.)		Bending Modulus of weakest section.
		At Journal	At Pin	At Journal	At Pin	At Journal	At Pin	
Hall Scott, 1917	1.00	.36	.36	.22	.22	.62	.62	.17
Sausie, 1921	1.00	.33	.33	.20	.20	.61	.61	
Liberty, 1918	.74	.23	.22	.14	.14	.40	.37	.10
Liberty, 1922	.74	.31	.30	.18	.17	.55	.60	
Liberty, 1923	.76	.31	.31	.18	.18	.55	.55	
Curtiss, 1917	1.00	.23	.23	.13	.13	.69	.69	.13
Curtiss, 1922	.58	.19	.19	.11	.11	.46	.46	
Curtiss, 1922	.58	.21	.20	.12	.12	.55	.51	
Lawrence, 1922-23	.82	.39 L .22 S	.39 L .22 S	.23 L .13 S	.23 L .13 S	.68 L .50 S	.68 L .50 S	.10 S
Van Blerck, 1922	1.00	.34	.34	.20	.20	.74	.74	
Sub.S-50, 1922	1.34	.63	.63	.39	.39	.97	.97	
Sub.S-1, 1913	.89	.55	.55	.35	.35	.82	.82	
USS PADUCAH, 1913	1.15	.68	.68	.47	.47	.83	.83	
USS C-4, 1912	.97	.67	.67	.42	.42	.97	.97	
Elec.Boat Co. S Type, 1921	1.00	.49	.49	.30	.30	.82	.82	
Subs.V-2 and V-3, 1922	1.00	.55	.55	.28	.28	.90	.90	

* S -Small, L - Large.

plane shafts than for the marine shafts. In many cases the ratios for the latter shafts are two or three times as large as for the former.

In Table IV are recorded nominal ratios of the stresses in various parts of each shaft to the stress in the journal. In making these calculations, it has been assumed that each shaft is subjected to uniform journal torque. In the columns headed "Restrained Journal" the effect of deflections in pin and webs on the torsional stress in the web has been neglected. The values in the last six columns of the table, therefore, are merely approximations. They are included here, however, to illustrate the fact that no elaborate calculation is necessary to show whether or not a crankshaft is so designed as to have prac-

tical uniformity of strength in its various parts. This table makes it evident that the older submarine shafts are somewhat more uniform in strength than the newer and are much more uniform in strength than the aeroplane shafts. In even the newest aeroplane shafts the webs are relatively weak in

TABLE IV.

Stress Distribution in Crankshafts.
Ratio of stresses in various sections to torsional stress in journal.*

Crankshaft & year received	Unrestrained Journal			Restrained Journal**					
	Torsional stress in Pin.	Bending Stress (Transverse) in web.		Torsional stress in web.		Bending stress (transverse) in web at jour.	Bending stress in Pin	Shear Stress (Average)	
		At Journal	At Pin	At Journal	At Pin			In Pin	In Web
Hall Scott, 1917	1.00	1.92	1.92	1.40	1.40	1.92	.71	.17	.20
Bessie, 1921	1.00	1.64	1.64	1.52	1.52	1.64	.71	.22	.25
Liberty, 1918	1.35	2.50	2.70	2.17	2.27	2.50	.97	.29	.34
Liberty, 1922	1.35	1.54	1.67	1.61	1.67	1.54	.97	.29	.27
Liberty, 1923	1.32	1.54	1.54	1.61	1.61	1.54	.95	.26	.26
Curtiss, 1917	1.00	1.45	1.45	3.37	3.37	1.45	1.32	.23	.29
Curtiss, 1922	1.72	2.17	2.17			2.17		.56	.43
Curtiss, 1922	1.72	1.89	1.96	2.36	2.48	1.89	1.23	.56	.39
Lawrence, 1922-23	1.22			2.19	2.19				
Van Blerck, 1922	1.00	1.35	1.35	1.93	1.93	1.35	1.00	.20	.28
Sub.S-50, 1922	.75	1.03	1.03	1.17	1.17	1.03	.69	.25	.25
USS G-1, 1913	1.12	1.22	1.22	1.18	1.18	1.22	.97	.19	.19
USS PADUCAH, 1913	.87	1.20	1.20			1.20			
USS C-4, 1912 Elec.Boat Co.	1.03	1.03	1.03	1.02	1.02	1.03	.98	.22	.16
S Type, 1921 1922	1.00	1.22	1.22	1.63	1.63	1.22	1.06	.30	.30
USS V-2 & V-3	1.00	1.11	1.11	1.69	1.69	1.11	1.38	.36	.27

* In making these calculations it is assumed that there is the same torsional moment in all pins and journals and that the bending moment in the webs is equal to the torsional moment in pins and journals.

** In obtaining these values no corrections have been made for the influence of strains of pin and web.

resistance to bending and especially in resistance to torsional stresses. It should be noted that in calculating web torsion ratio use was made of the formula developed above.

Using the stress ratio given in Table IV calculation has been made of the deflections in each shaft due to 1,000 pounds inch torque in the journals and the results are given in Table V. For example, the values given in Columns 4 and 5 are the distances by which two adjacent journals are moved out of line due to twist in the pin and bending deflection of one web,

respectively. It will be noted that the values given in Column 6 are half the values given in Column 5. This is due to the fact that in the restrained shaft the effect of torque in the pin on bending the web is lacking. It is obvious that in making these calculations for deflection the values of the equivalent length of each part of the shaft must be arbitrarily assumed.

TABLE V.

Deflections and displacements in crankshafts, under torque, corresponding to Torsional moment of 1000 in.lb. in journal.

Crankshaft and year received.	Unrestrained Journals. Twist in Journal Radian	Twist in Pin Radian	Displacement of Journal due to Twist in pin, in.	Displacement of Journal due to Bending of web, in.	Restrained Journals. Displacement of journals due to Bending of web, in.	Twist in web, in.	Bending of pin, in.
Hall Scott, 1917	.000102	.000127	.000445	.000245	.000123	.000426	.0000175
Hausie, 1921		.0000959	.000259	.000106	.0000531	.000207	.0000100
Liberty, 1918	.0000357	.0000567	.000233	.000126	.0000634	.000191	.00000923
Liberty, 1922	.0000357	.0000567	.000233	.0000613	.0000306	.000141	.00000923
Liberty, 1923	.0000348	.0000636	.000223	.0000550	.0000275	.000135	.00000880
Curtiss, 1917	.000123	.000224	.000560	.0000882	.0000441	.000926	.0000760
Curtiss, 1922				.0000555	.0000277		
Curtiss, 1922	.0000274	.0000628	.000188	.0000471	.0000236	.0000467	.00000741
Lawrence, 1922-23	.000247						
Van Blerck, 1922	.0000612	.0000841	.000231	.0000481	.0000241	.000146	.0000179
Sub.S-50, 1922	.00000809	.00000108	.00000884	.00000243	.00000122	*	.000000594
Sub.C-1, 1913	.00000890	.00000924	.0000641	.0000137	.00000685	.0000221	.00000371
USS PADUCAH, 1913							
Sub.C-4, 1912	.0000149	.0000155	.0000940	.0000166	.0000083	.0000373	.00000632
Elec.Boat Co. S Type, 1921	.00000156	.00000156	.0000117	.00000227	.00000113	*	.00000103
Sub.V-2 & V-3 1922	.00000104	.00000117	.00000969	.00000127	.000000635	*	.00000132

* Since pin and journal sections overlap on opposite faces of the web, no attempt was made to calculate the displacement of journal due to torsion in web.

The influence of abrupt changes of section and of direction cannot be exactly determined. It is believed, however, that the results given in this table represent approximate relationships between the actual deflections of the various parts.

Using the values given in Table V calculation has been made of the stresses in the various parts of each shaft corresponding to a journal torque of 1,000 pounds inch. The results are given in Table VI-C. For comparison, the stresses in the free shaft and the uncorrected stresses in the restrained shaft

have been calculated using the stress ratios given in Table IV. The results are given in Tables VI-A and VI-B, respectively.

In order to make allowance for the influence of deflections in the pin and webs of the restrained shaft, it is necessary to calculate a fraction K by means of which values as given in some of the columns of Table VI-C may be obtained, and if desired the ratios given in some of the columns of Table IV, may be corrected. In a previous paragraph formulas were

TABLE VI. (A)

Stress Distribution in Crankshafts under 1000 lb. in. Journal Torque. Stress in lb. per sq. in.

Crankshaft and year received.	Unrestrained Journals.			
	Torsional stress in Journal	Torsional stress in Pin	Bending stress, (Transverse) in web.	
			At Journal	At Pin.
Hall Scott, 1917	662	662	1272	1272
Hausie, 1921	559	559	916	916
Liberty, 1918	305	412	762	823
Liberty, 1922	305	412	470	509
Liberty, 1923	298	393	458	458
Curtiss, 1917	826	826	1198	1198
Curtiss, 1922	276	475	599	599
Curtiss, 1922	276	475	522	541
Van Blerck, 1922	448	448	605	605
Sub.S-50, 1922	10.8	8.1	11.1	11.1
Sub.G-1, 1913	42.4	47.5	51.7	51.7
USS PADUCAH, 1913	40.8	35.5	49.0	49.0
USS G-4, 1918	70.7	72.8	72.8	72.8
Electric Boat Co. S Type, 1921	10.8	10.2	12.4	12.4
Sub.V-2 & V-3, 1922	6.51	6.51	7.23	7.23

developed to represent the conditions existing in a restrained shaft assuming the stiffness of the pin and webs to be so great that deflection would not influence stress distribution. The force with which a journal is laterally restrained is represented by F. Actually, however, torsional deflection of the webs would result in diminution of this force and in retention of some torque in the pin. The actual force with which a journal is laterally restrained may be represented by KF. The residual torque in the pin would then be (1-K)M.

The value of K may be determined by forming an equation in which the algebraic sums of the lateral displacements of two adjacent journals, due to deflections of the pin and webs, are made equal to zero. This would be the case if the two journals

TABLE VI. (B)

Stress Distribution in Crankshafts under 1000 lb. in. Journal Torque. Stress in lb. per sq. in.

Crankshaft and year received.	Restrained Journal. Uncorrected for influence of strains of Pin & Web. Bending Stress		Torsional stress in web		Shearing Stress (Average)	
	Pin	Web (Transverse) At Journal	At Journal	At Pin	Pin	Web
Hall Scott, 1917	472	1272	927	927	112	135
Emsie, 1921	395	916	849	849	121	138
Liberty, 1918	295	762	662	692	89.3	104
Liberty, 1922	295	470	491	509	89.3	81.6
Liberty, 1923	281	458	479	479	78.5	78.7
Curtiss, 1917	1091	1198	2785	2785	195	237
Curtiss, 1922		599			156	117
Curtiss, 1922	339	522	652	695	156	109
Van Blerck, 1922	446	606	865	865	91.4	123
Sub. S-50, 1922	7.61	11.1	12.7	12.7	2.67	2.54
Sub. O-1, 1913	41.1	51.7	48.7	48.7	8.13	8.06
USS PADUCAH, 1913		50.0				
USS C-4, 1912	69.4	72.8	72.1	72.1	15.5	11.5
Electric Boat Co. S Type, 1921	10.8	12.4	16.6	16.6	3.09	3.03
Sub.V-2 & V-3, 1922	8.63	7.23	11.0	11.0	2.35	1.77

are held rigidly in line by their bearings. On this principle the following equation is formed:—

2 [(restrained web bend) K + (twice restrained web bend) (1-K)] + (free pin twist) (1-K) − (web twist) K − (pin bend) K = 0. Using this equation and substituting the values given in Table V, the values for K given in Table VI-C have been obtained. Using these values of K as described in the note at the end of Table VI-C, the corrected values in other columns of this table have been obtained.

A comparison of the columns numbered 2 and 5 shows that in nearly all these shafts the transverse bending stress in the web is greater than the torsional stress calculated for the smallest perpendicular cross section. In addition to the shear-

ing stress in the web due to torsion, however, there is shearing stress in the entire cross section of the web. The values of this average shear stress for the various shafts are given in Column 7 of Table VI-C. At the re-entrant angle between web and pin this shearing stress in the web is about 50 per cent greater than the values given in Column 7 and is in the

TABLE VI. (C)

Stress Distribution in Crankshafts under 1000 lb. in. Journal Torque.
Stress in lb. per sq. in.

	(1)	(2)		(3)	(4)	(5)	(6)	(7)
	Reversed Bending Stress. Corrected for influence of strains of pin and web.			Torsional Stress in Web		Torsional stress in Pin	Shearing Stress, (Average)	
Crankshaft and year received.	"K"	Pin	Web (transverse) at Journal	At Journal	At Pin		Pin	Web
Hall Scott, 1917	.85	392	1272	770	770	115	93.5	112
Hauele, 1921	.81	380	916	686	688	106	97.8	112
Liberty, 1918	.87	257	762	576	602	83.5	77.7	90.4
Liberty, 1922	.80	236	470	393	407	82.4	71.4	65.3
Liberty, 1923	.79	222	458	378	378	82.5	62.0	62.2
Curtiss, 1917	.45	491	1198	1288	1253	454	86.9	106
Curtiss, 1922			599					
Curtiss, 1922	.98	353	522	639	671	9.80	183	107
Van Blerck, 1922	.74	330	605	640	640	117	67.6	91.8
Sub.S-50, 1922			11.1					
Sub.C-1, 1915	.88	36.2	51.7	42.9	42.9	5.70	7.15	7.09
USS PADUCAH, 1915			49.0					
USS C-4, 1912 Electric Boat Co. S Type, 1921 Subs V-2 & V-3, 1922.	.82	56.9	72.8	59.1	59.1	13.1	12.7	9.48
			12.4					
			7.23					

"K" = Factor by which various nominal stress values must be multiplied to allow for influence of strain in web and pin on stress distribution. The value of "K" is obtained from the following equation:-

$$\Sigma\left[(\text{web bend})K+2(\text{web bend})(1-K)\right] - (\text{web twist})K-(\text{pin bend})K_1+(\text{pin twist})(1-K) = 0.$$

NOTE:- Columns 1, 3, 4, 5 and 7 obtained by multiplying the corresponding nominal values in Table VI-B by "K".
Column 2 has no correction.
Column 6 is obtained by multiplying the corresponding nominal value in Table VI-A by (1-K).

same direction as the shearing stress due to torsion. In the pin, however, the torsional stress and the additional shearing stress are in opposite directions. The values for the combined stress due to torsion and shear in web and pin are given in Table VI-D.

At the re-entrant angle the metal is subject to shearing stresses of both web and pin acting in directions perpendicular

to each other. The resultants of these pin and web stresses at the re-entrant angle may be calculated using the formula

$$Sr = \sqrt{S_w^2 + S_p^2}$$

The resultant values are given in the last column of Table VI-D. They differ only slightly from the values given in the first column. It should be noted, however, that if there should

TABLE VI. (D)

Stress Distribution in Crankshafts under 1000 lb. in. Journal Torque. Stress in lb. per sq. in.

Crankshaft and year received.	Combined stress due to Shear & Torsion. In web at reentrant angle between web and pin. * Sw.	Combined stress due to Shear & Torsion. In pin at reentrant angle between pin and web. ** Sp.	Final Resultant Shearing stress at reentrant angle between web and pin. *** Sr.
Hall Scott, 1917	938	27.4	939
Rausie, 1921	856	40.5	857
Liberty, 1918	738	63.1	741
Liberty, 1922	505	24.7	506
Liberty, 1923	472	11.5	472
Curtiss, 1917	1413	324	1450
Curtiss, 1922			
Curtiss, 1922	831	220	860
Van Blerck, 1922	777	15.2	777
Sub.S-50, 1922			
Sub.G-1, 1913	53.5	5.03	53.8
USS PADUCAH, 1913			
USS G-4, 1912 Electric Boat Co. S Type, 1921 Subs. V-2 & V-3 1922	73.3	5.98	73.5

* Sw = The sum of corrected torsional stress in web (at pin) and 1.5 x the average corrected shearing stress in web.

** Sp = The difference between the residual torsional stress in pin and 1.5 x the corrected shearing stress in pin.

*** $Sr = \sqrt{(Sw)^2 + (Sp)^2}$

be any yield in the bearings tending to decrease the web torque and increase the pin torque the resultant stress at the re-entrant angle would remain considerably higher than that due to either web torque or pin torque alone.

In Table VII is given a comparison between the stresses in the journal, at the edges of the web, and at the re-entrant angle between web and pin. A comparison of the values given in the

FIG. 13.

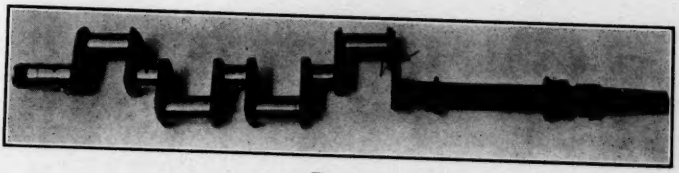


FIG. 14.

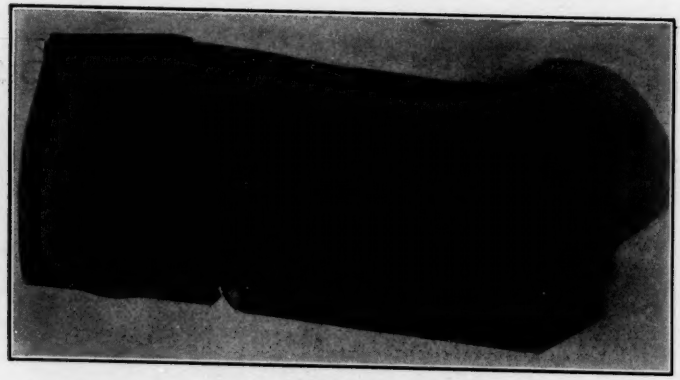


FIG. 15.

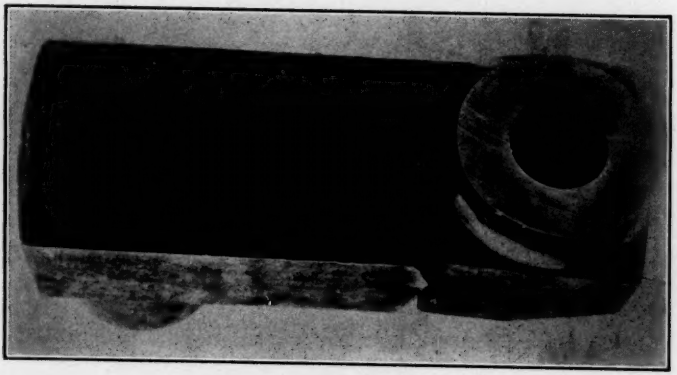


FIG. 16.

last column of Table VII with the ratios given in the fifth column of figures in Table IV shows that with the exception of the values for Curtiss shafts, the nominal values recorded in Table IV agree very well with the corrected values recorded in Table VII. With few exceptions, the stress at the re-entrant angle is higher than the stress at the edge of the web due to bending.

TABLE VII.

Comparison of bending stress in web with torsional stress in Journal and probable torsional stress at reentrant angle between web and pin.
Torsional stress in journal 1000 lb. per sq. in.

Crankshaft and year received.	Bending stress (Tragsversa) in web at journal lb. per sq.in.	Final Resultant shearing stress at reentrant angle between pin & web. lb. per sq. in.
Hall Scott, 1917	1920	1417
Hausie, 1921	1640	1534
Liberty, 1918	2500	2489
Liberty, 1922	1540	1660
Liberty, 1923	1540	1586
Curtiss, 1917	1450	1755
Curtiss, 1922	2170	-
Curtiss, 1922	1890	2115
Van Blerck, 1922	1350	1734
Sub.S-50, 1922	1030	* 1170
Sub.O-1, 1918	1220	1150
USS PADUCAH, 1918	1300	
USS C-4, 1918	1030	1040
Electric Boat Co. S Type 1921	1220	* 1650
Subs.V-2 & V-3, 1922	1110	* 1690

* This value is merely nominal due to the fact that strain in pin and web cannot be calculated since journal and pin overlap. Since strains in web are very small and since the overlaping is slight, it is believed that this nominal value is not far from the actual value.

It should be noted, moreover, that the stresses at the re-entrant angle are calculated for the smallest perpendicular cross section. This is not the weakest section in a crankshaft, having lengthwise holes in journals and pins. As previously stated, the fracture usually extends from the re-entrant angle diagonally through the web into the axial oil hole in pin or journal. Views of such fractures are shown in Figs. 10 to 13, inclusive, 15 and 16. Since in a rectangular section under torque the stress is highest at the middle of the longest side

of the rectangle, it is evident that these sections of fracture (in which the middle portion of one side of the rectangle is missing) are especially weak in resistance to torsion. They are also decidedly weak in resistance to longitudinal bending while their resistance to transverse bending is not much less than that of other transverse sections of the web. It seems evident, therefore, that in these sections the stress ratios were actually much higher than those indicated in the last column of Table VII or in the fifth column of figures of Table IV.

In order that failure due to torque in the shaft may occur through the web rather than through the journal, the stress at the re-entrant angle must evidently be not merely greater than the nominal stress in the journal, but also greater than the localized stress at the radial oil holes in the journal. The stress at such radial holes in a shaft of heat-treated alloy steel may be about twice as great as the stress elsewhere on the surface of the journal. In a shaft of softer steel, the stress magnification is probably somewhat less. It seems extremely probable, however, that in the diagonal section through which crank web failures occur the stress is actually enough greater than the values given in the last column of Table VII to offset the stress localization at the radial oil holes in the journal.

MEASURED AND CALCULATED ANGULAR DEFLECTIONS OF LIBERTY CRANKSHAFT.

Some results of angular deflection measurements have been sent to the Experiment Station in support of the theory that there is no important lateral restraint of journals and hence no important web torque in crankshafts. These measurements were made on a Liberty crankshaft of the same type as that designated in Tables I to VII as Liberty, 1922. These results indicate that for the free shaft the deflection rate is 0.081 degrees per thousand pounds inch torque while for the shaft in its crankcase the deflection rate is 0.072 degrees. From these results it was argued that any restraint due to the bearings has no important effect on stress distribution in the shaft.

Calculation of stress-strain relationships for free and for restrained shaft using the data given in Table VI shows, however, that the angular deflection measurements do not disprove the existence of web torque in this shaft.

The angular deflection per thousand pounds inch torque for free and for restrained shaft may be calculated using values given in Table V. Evidently in the journals the angular deflection would be the same whether the shaft is free or restrained. In the webs and pins, however, the stress distribution, and hence the deflections, would be entirely different under the two different conditions. To compare those of the free with those of the restrained shaft, a comparison of the angular deflections of the shaft due to deflections of a single pin and its two adjacent webs should give information of value.

For the Liberty shaft, with journals free the angular deflection in radians due to the bending deflection of a web=MR/EI. Since in Table V the value given in the fourth vertical column of figures was obtained by use of the formula $MR^2/2EI$, that value must be doubled and divided by R to give the angular deflection in radians. The value thus obtained is .000035. For both webs the value is .00007. Adding the value for pin twist given in the second column of figures, we obtain 0.000137 radians for the angular deflection of the free shaft due to deflections of a pin and its two adjacent webs.

For a shaft with laterally restrained journals the angular deflection due to deflections in a pin and two adjacent webs may be expressed in the same terms that were used in a previous paragraph. Using these terms it can be shown that the angular deflection of the shaft in radians due to deflection of these parts is equal to $\frac{(8-4K)}{R}$ times (Displacement of journal due to restrained web bend)+(1-K) times (Unrestrained pin twist) $+\frac{K}{R}$ times (Displacement of journal due

to web twist and pin bend.) Substituting the values given in Table VI, the formula becomes $0.0000306 \frac{(8-4K)}{R} + 0.0000667\ (1-K) + (0.000141 + 0.0000092) \frac{K}{R}$ which gives 0.000099 radians for the angular deflection of the restrained shaft due to deflections in a pin and its adjacent webs.

A comparison of the values obtained in the two preceding paragraphs indicates that the angular shaft deflection due to deflections in a pin and its adjacent webs is about 28 per cent less in the restrained than in the free shaft. For the entire shaft the percentage difference would evidently be considerably less, since in calculating the total angular deflections the journal deflections must be included and since these deflections are the same in the restrained as in the free shaft. In the measurements as received, the twisted portion of the shaft extended from the propeller end to the pin adjacent to the anti-propeller end. If the angular deflections for this portion of the shaft are calculated according to the principles described in the two preceding paragraphs using deflection values given in Table V, the values obtained for free and for restrained shaft are 0.00096 radians and 0.00078 radians respectively. The calculated angular deflection for the restrained shaft is, therefore, 18.8 per cent less than for the free shaft.

These calculations have served to show, therefore, that restraint of journals and consequent high web torque are possible even if the angular deflection of the restrained shaft is only slightly less than that of the free shaft. Although the calculated percentage difference is somewhat greater than the measured percentage difference it is belived that little significance should be attached to this slight discrepancy. Especially in view of the fact that the calculated deflection of the free shaft is so much less than the measured deflection of the same shaft.

SOME TYPICAL CRANKSHAFT FAILURES.

Not only do theoretical considerations lead to the conclusion that considerable torque may exist in the webs of crankshafts, but also information obtained from a study of crankshaft failures. It would be highly desirable to include photographs of the many fractures that were sent to the Experiment Station from time to time. However, since space is limited, a few typical examples must suffice.

In Fig. 10 is shown a failed Hall Scott crankshaft. This shaft failed through a web at the junction with a journal. Since the journals and pins are of the same size, a web section at its junction with a journal is no larger than at the junction with a pin. The fracture extends partly around the journal and then straight across the web through the axial oil hole in the journal. The appearance of the fracture seems to indicate that this fracture occurred chiefly under the influence of transverse bending stresses in the web at its weakest section. The general appearance of this shaft as compared with those of more modern design would indicate that the web breadth is much too small. As shown in Table VII, the transverse bending stress in the web is considerably greater than the resultant shearing stress at the reentrant angle between web and journal. The calculated bending stress is 1,920 pounds per square inch while the torsional stress in the journal and the resultant shearing stress at the re-entrant angle are 1,000 and 1,417 pounds per square inch respectively. Although in a diagonal section from the re-entrant angle to axial oil hole the resultant shearing stress is undoubtedly considerably greater than the nominal value given in Table VII, the bending stress in the web was probably also an important factor in the failure of this shaft.

In Fig. 11 is shown a failed Rausie crankshaft. Unfortunately no enlarged view of the surface of fracture is available. It is possible that this is the same type of fracture as that of the Hall-Scott shaft described above. A reference to Table VII will show that the transverse bending stress in the web

is somewhat higher than the resultant shearing stress at the re-entrant angle, and that these stresses are much higher than the torsional stress in the journal. In the section of actual fracture it is possible that the resultant shearing stress was higher than the bending stress.

In Fig. 12 is shown a failed Liberty crankshaft of old design. The fracture is in the weakest section extending from a reentrant angle through the web into the axial hole in the pin. On account of the distortion of the metal in the fracture it is impossible to decide from the appearance of the fracture just where the crack started. As shown in Table VII in the third space from the top, the nominal bending stress is slightly greater than the nominal resultant shearing stress in these webs. At the weakest section, however, the torsional stress would be increased much more than would the transverse bending stress above the nominal values given in Table VII. Assuming that this fracture was due to critical torsional vibrations and not to longitudinal bending, this failure could, therefore have been due to either transverse bending or resultant shear in the web. It should be noted that in this shaft as indicated in Table VII, each kind of stress in the web is about 2½ times as great as the torsional stress in the journal.

In Fig. 13 is shown a failed Liberty shaft of more recent design than that shown in Fig. 12. The newer shaft differs from the other shaft in that the web section has been considerably increased. As shown in Fig. 13 the fracture extends from the re-entrant angle through the web and into the axial hole in the pin. In this shaft as shown in Table VII the nominal resultant shearing stress at the reentrant angle is greater than the nominal transverse bending stress in the web. In the weakest section, through which fracture occurred, the resultant shearing stress was undoubtedly much greater than the transverse bending stress. In order to account for failure through this region rather than through a radial oil hole in the journal, the actual stress at the re-entrant angle must be assumed to be at least twice the nominal surface stress in the journal.

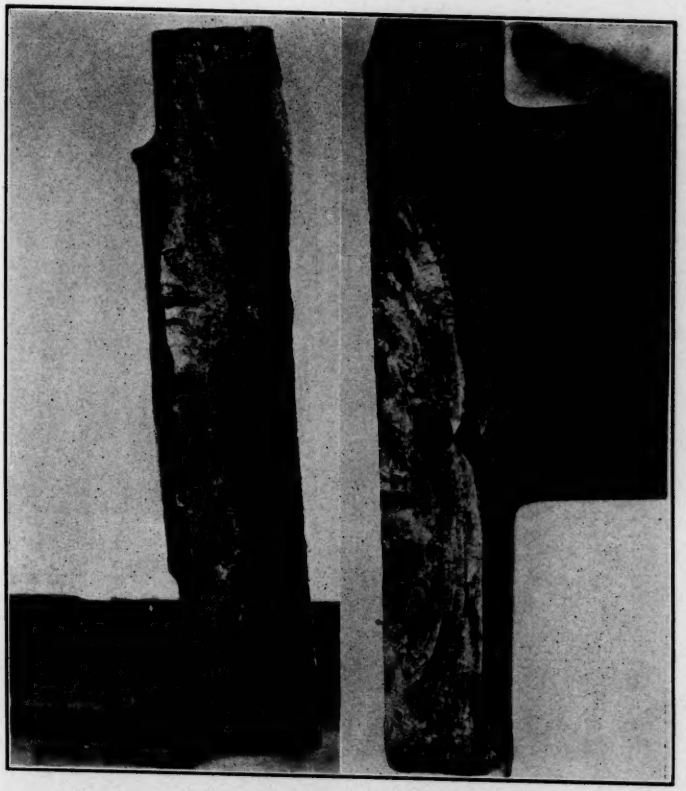


FIG. 17.

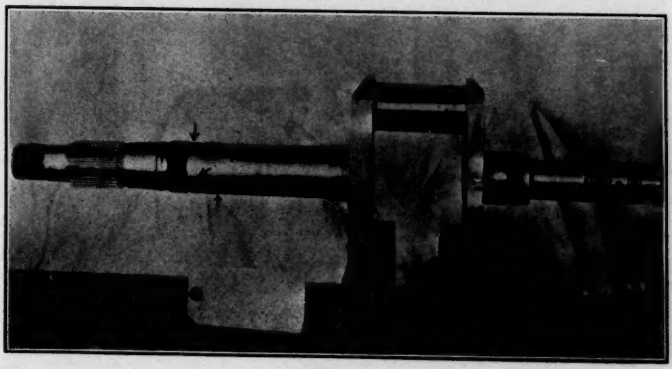


FIG. 18.

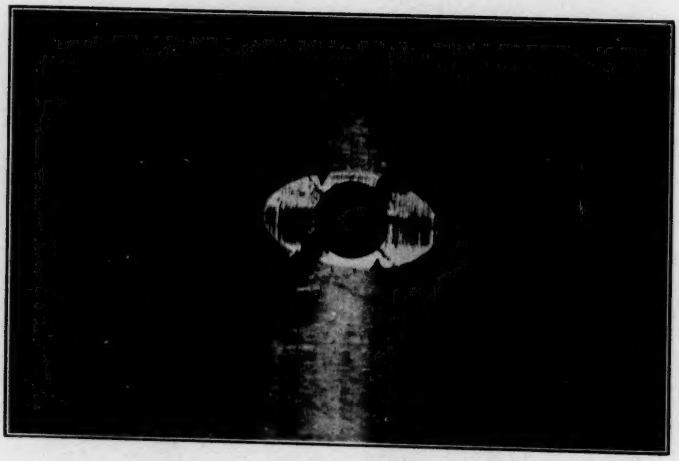


FIG. 19.

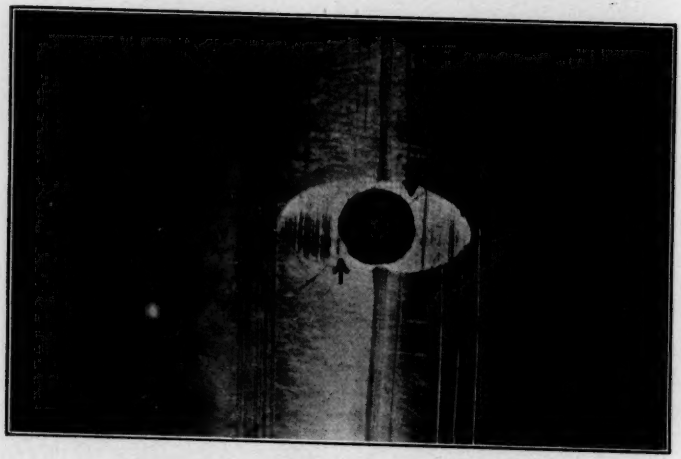


FIG. 20.

Figs. 1, 14 and 17 show a failed Curtiss shaft of old design. Examination of this fracture shows beyond doubt that it started at the re-entrant angle between a web and pin. In Fig. 1 is shown a crack in another part of this shaft. If the shaft had not broken elsewhere, this crack would soon have advanced through the web and caused complete fracture. Since in these shafts the web was so thin the fractures proceeding diagonally traversed the web before reaching the axial hole in the pin.

As shown in Table VII in these shafts (Curtiss, 1917) the nominal resultant shearing stress at the re-entrant angle is much greater than the nominal transverse bending stress in the web, while each of these stresses are considerably greater than the torsional stress in the journal.

The type of fracture shown in Fig. 17 could have been due to longitudinal bending of the web. They could also have been due chiefly to torsional stress in the web caused by critical torsional vibration of the shaft. The latter alternative seems the more probable.

Figs. 18 to 20 inclusive, show another failed Lawrence J-1 shaft. The arrows in Fig. 18 point to part of the spiral crack in the journal. This crack evidently started at an oil hole in the journal and extended in opposite directions in a spiral making an angle of 45 degrees with the shaft axis. The fact that this crack started at an oil hole was made evident by a closer examination of the shaft. By magnetizing the shaft and flowing over it a mixture of iron filings and kerosene, the filings adhered to the shaft so as to mark the location of other incipient cracks. In Fig. 19 are shown these incipient cracks indicated by arrows. As here shown incipient cracks have started from the same oil hole from which the large crack started. These cracks are at right angles to the large crack thus indicating that the shaft was subjected to alternating torsional stresses. As shown in Fig. 20 incipient cracks have started from another oil hole in the same spiral direction as the large crack.

Evidently, therefore, this shaft had been subjected to alternating torsional stresses due to critical torsional vibrations. The localized torsional stresses around the oil holes in this kind of steel as shown by experiments at this Station, are about twice the nominal torsional stresses in the journal. Since fracture started at this point the localized stresses were evidently greater than stresses elsewhere in the shaft. It should be mentioned that in this single throw shaft the web stress distribution is entirely different from that found in crankshafts of the types previously discussed. In this shaft, restraint of journals is not an important factor in producing web torque.

Fig. 21 shows the fracture of a failed shaft of a Van Blerck engine. This shaft failed through a web at its junction with the journal. Examination of the fracture shows unmistakably that the crack started at the re-entrant angle. From this point the fracture radiated and the segment of a circle thus formed had extended more than half way through the web thickness before it reached the web edges. This fracture, therefore, was due chiefly either to longitudinal web bending or to web torque. It could not have been due to transverse web bending.

As shown in Table VII the resultant shearing stress at the junction of web and pin was considerably greater than the transverse bending stress in the web and was 75 per cent greater than the torsional stress in the journal. Evidently, however, the actual stress at the re-entrant angle must have been at least twice the nominal journal stress, otherwise the shaft would have failed through a radial oil hole in a journal. It can be shown that in this shaft the resultant stress at the junction of a web and journal is considerably greater than the resultant stress at the junction of a web and pin, as recorded in Table VII. This is due to the fact that in the restrained shaft there is reduced torque in the pin while the journal is subjected to full torque. The resultant stress at the junction of web and journal is, therefore, probably about $\sqrt{1{,}000^2 + 1{,}734^2} = 2{,}000$ when the nominal journal stress is 1,000 pounds per square

FIG. 21.

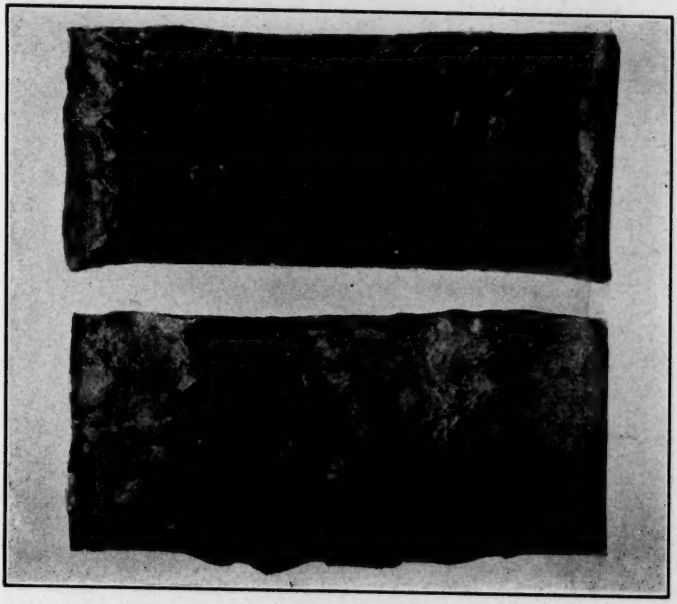


FIG. 25

FIG. 22.

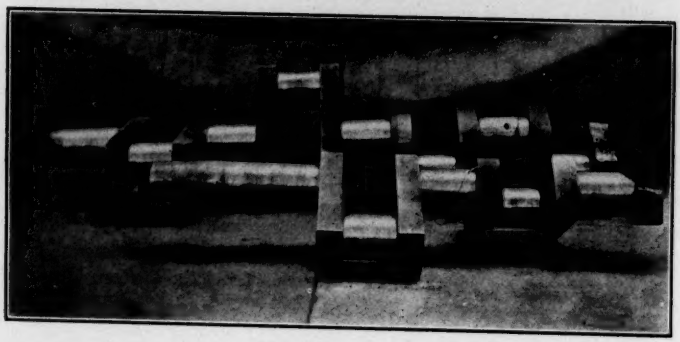


FIG. 24.

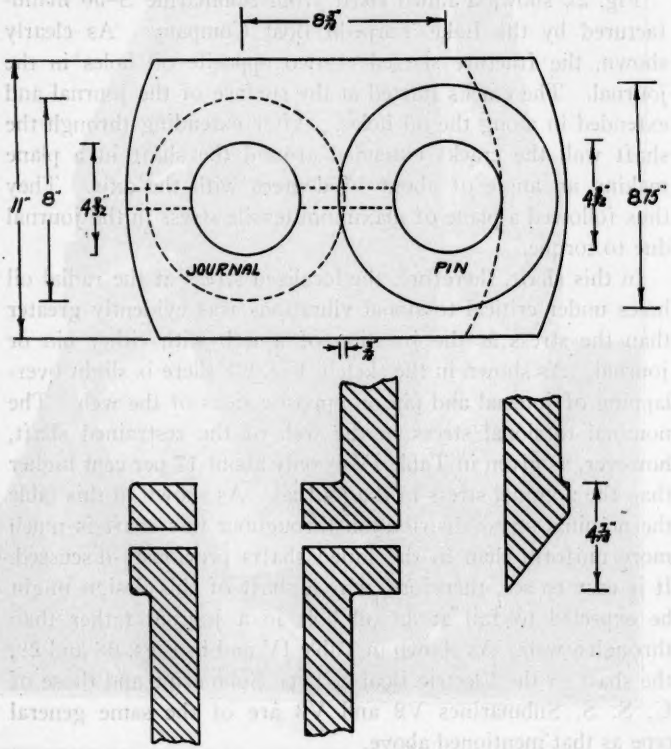
inch. This calculated resultant stress at this junction is, therefore, sufficient to account for the failure of the shaft at this point.

Fig. 22 shows a failed shaft from Submarine S–50 manufactured by the Lake Torpedo Boat Company. As clearly shown, the fracture started at two opposite oil holes in the journal. The cracks started at the surface of the journal and extended in along the oil holes. After extending through the shaft wall the cracks extended around the shaft in a plane making an angle of about 45 degrees with the axis. They thus followed a plane of maximum tensile stress in the journal due to torque.

In this shaft, therefore, the localized stress at the radial oil holes under critical torsional vibrations was evidently greater than the stress at the junction of a web with either pin or journal. As shown in the sketch, Fig. 23, there is slight overlapping of journal and pin on opposite sides of the web. The nominal torsional stress in the web of the restrained shaft, however, as given in Table IV is only about 17 per cent higher than the nominal stress in the journal. As shown in this table the nominal stress distribution throughout this shaft is much more uniform than in the failed shafts previously discussed. It is easy to see, therefore, why a shaft of this design might be expected to fail at an oil hole in a journal rather than through a web. As shown in Table IV and in Figs. 28 and 29, the shaft of the Electric Boat S-Type Submarine and those of U. S. S. Submarines V2 and V3 are of the same general type as that mentioned above.

Figs. 24 and 25 show the failed starboard shaft of Submarine G-1. The failed port Shaft of this submarine was also sent to the Experiment Station. The history shows that on one of the shafts longitudinal cracks in several pins had been found after about 40 hours run by the Navy. It was also found that the bed plates were out of line and the shafts were "out of round." With the information now available this would have led to the conclusion that this shaft had been subjected

to critical torsional vibrations of sufficient degree to cause lateral displacement of journals. The interpretation of the cracks given by the board of investigation was that they were



JOURNAL WEB AND PIN SECTION SUB S-50

FIG. 23.

the extension of the central ingot pipe remaining in the slab forged shaft. This may have been the case. Such a region of weakness would serve as a starting point for the failure under alternating torsional stress due to critical vibrations.

FIG. 26.

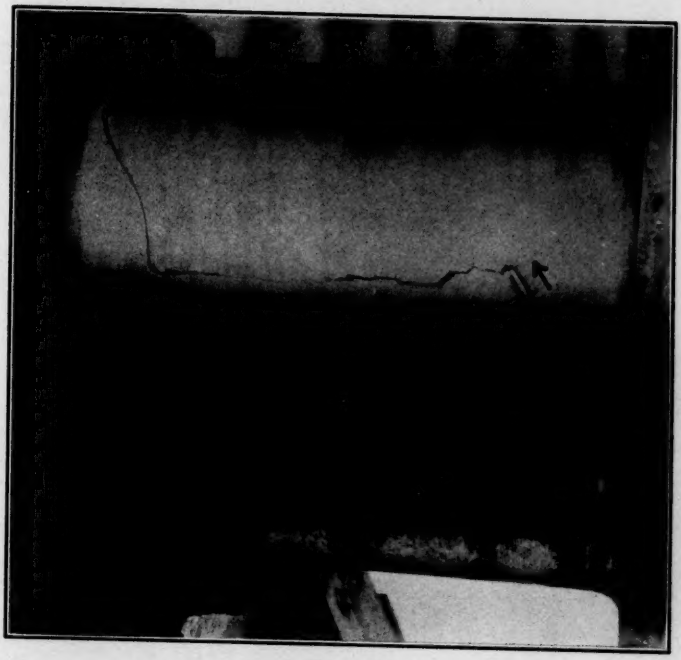
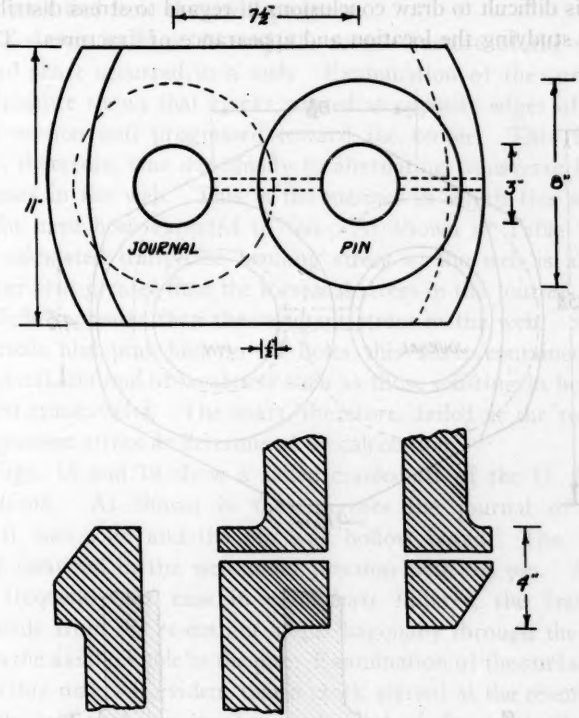


FIG. 27.

These failed shafts had numerous cracks on pins and journals. Undoubtedly there were also numerous cracks in the webs. These would not be so easy to find as those on pins



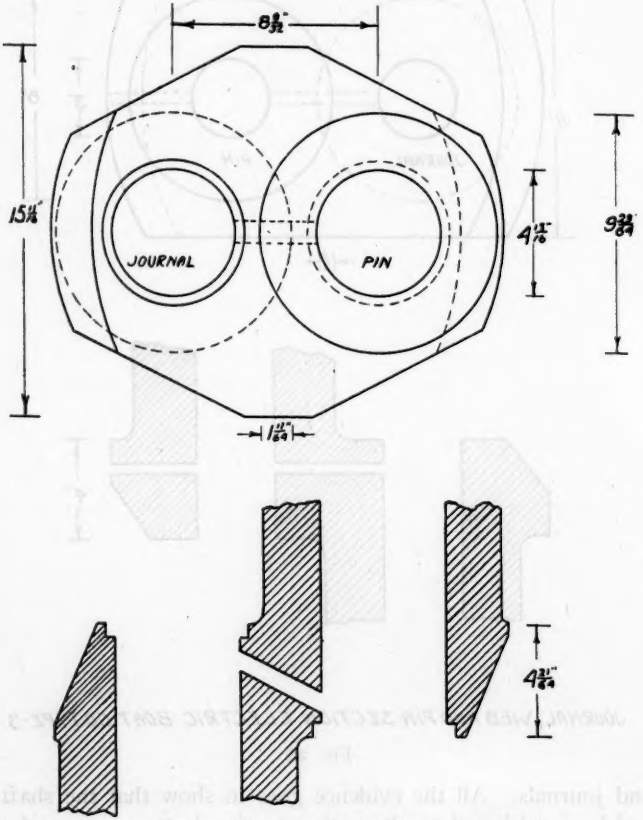
JOURNAL, WEB AND PIN SECTION ELECTRIC BOAT CO TYPE-S

FIG. 28.

and journals. All the evidence goes to show that the shafts had been subjected to alternating torsional stresses to such a degree that cracks had started in all parts of each shaft.

The metal in these shafts was undoubtedly defective. It contained a large quantity of non-metallic inclusions segre-

gated in longitudinal streaks. There was also streaked segregation of carbon. The cracks in the shaft started at these regions of segregated inclusions. In such defective material it is difficult to draw conclusions in regard to stress distribution by studying the location and appearance of fractures. This is



JOURNAL, WEB AND PIN SECTION SUBS V2 AND V3

FIG. 29.

due to the fact that in defective material the regions of weakness, due to localized defects rather than the regions of highest stress, may be the chief factor in determining where fractures will start and how they will progress.

As shown in Figs. 24 and 25, actual fracture of the starboard shaft occurred in a web. Examination of the surface of fracture shows that cracks started at opposite edges of the web section and progressed toward the center. This fracture, therefore, was due chiefly to alternating transverse bend stresses in the web. This is the manner in which this shaft might have been expected to fail. As shown in Table VII, the calculated transverse bending stress in the web is about 35 per cent greater than the torsional stress in the journal, and is slightly greater than the resultant stress in the web. Since journals and pins had no oil holes this shaft contained no diagonal sections of weakness such as those existing in hollow bored crankshafts. The shaft, therefore, failed at the region of greatest stress as determined by calculation.

Figs. 15 and 16 show a failed crankshaft of the U. S. S. *Paducah*. As shown in these figures the journal of this shaft was solid and the pin was hollow bored. The fracture occurred in the web at its junction with the pin. As is so frequently the case in crankshaft failures, the fracture extends from the re-entrant angle diagonally through the web into the axial oil hole in the pin. Examination of the surface of fracture makes it evident that a crack started at the re-entrant angle, and probably another at the end of the axial oil hole. Unfortunately the actual dimensions of the web are not available. The general appearance as shown in the Figure would indicate that the calculated torsional stress in the web would probably have been higher than that in the journal. The appearance of the fracture, however, leaves no doubt that the failure was due to either web torque or longitudinal web bending.

Fig. 26 shows a section of the failed shaft of Submarine K-6. As there shown, the fracture occurred in a journal. It

started at an oil hole and followed at first a longitudinal direction, the direction of least resistance to shear. Unfortunately the dimensions of the webs were not measured before the material was cut up. It seems probable that calculations would show that the web stress was less than twice the nominal stress in the journal. If that was the case it might have been expected that failure would occur as it did.

Fig. 27 shows a failed shaft of Submarine K-7. As shown in this figure, this fracture started as a longitudinal crack in a journal. This crack branched in two directions at 45 degrees at the ends of the longitudinal crack (as can be seen from the photograph), one of these branches continuing spirally around the shaft. The fact that this crack did not start at an oil hole would seem to indicate that a crack in the forging was the chief factor in determining the location of the failure of the finished shaft. The type of crack shown in Fig. 27 has been frequently noticed at this Station in alternating torsion specimens that failed under fatigue. This failure was undoubtedly due to alternating torsional stresses caused by critical vibrations.

A study of the crankshaft failures, described above, seems to substantiate the conclusions reached as a result of theoretical considerations. In aeroplane crankshafts, except the single throw Lawrence shaft, the weakest section is at the re-entrant angle between a web and either pin or journal. In all except shafts of old design the character of the fracture is such that it can be due chiefly either to web torque or to bending of the web in the direction of the shaft axis. In a few failed shafts of old design, transverse bending of the web may have been an important factor in causing failure. In these old shafts, however, the calculated transverse bending stresses in the web are higher than the calculated resultant shear stresses. In all the newer shafts the converse is true.

In Marine and Submarine crankshafts, with the exception of some shafts of old design, such as those of the *Paducah* and G-1, failure tends to occur in a journal, usually at a radial oil

hole. This location of failure is in agreement with the conclusions reached by calculation, that in these shafts, the localized stress at the oil hole is greater than the stress at junction of a web with either pin or journal.

DESIRABLE STRESS DISTRIBUTION IN CRANKSHAFTS.

If a crankshaft is run at a speed of critical torsional vibrations, it will probably fail sooner or later. The failure will usually occur in the region of highest calculated stress. The location of this region is of minor importance, provided the shaft has been so designed that the stress throughout under these conditions is fairly uniform. If, however, the variation of stress throughout the shaft is great, an effort should be made to redesign the shaft so as to render the stress distribution more uniform.

As shown in Table VI in aeroplane crankshafts the nominal stresses throughout are far from uniform. The webs in aeroplane shafts are much weaker relative to the journals than in most marine or submarine shafts. In order to minimize length, the webs have been made thinner than should be the case if uniformity is desired. For this reason, aeroplane shafts are especially weak in their resistance to either web torque or longitudinal web bending. Since for either longitudinal web bending or web torque the section modulus is proportional to Bb^2, in which B and b are web breadth and web thickness respectively, the ideal way to increase the relative strength of the web would be to increase the web thickness. If for any reason this cannot be done, much may be accomplished by increasing web breadth. This, however, would increase transverse bending strength more than it would increase longitudinal bending strength or torsional strength.

It should be noted that, while, as shown in Table IV, the Liberty shaft with each redesigning has been tending toward greater uniformity of strength, the Curtiss shaft has recently tended toward less uniformity. This is due to the fact that in the Curtiss shaft the journal diameter has been greatly

increased without corresponding increase of the dimensions of other parts of the shaft. The great relative size of the journals, while it gives greater bearing surface, has not made the new Curtiss shaft as a whole any stiffer in resistance to torque than is the new Liberty shaft. The calculated angle of twist for each entire shaft under the same torque is practically the same.

In marine and submarine shafts, with the exception of the Van Blerck, the nominal stress distribution as shown in Table IV is much more uniform than in aeroplane shafts. In some of the newer submarine shafts, however, the resistance of the web to torsion or longitudinal bending is relatively less than in older submarine shafts. This is due to the fact that the web thickness has not been increased to the same extent as has the web breadth.

In Table IV the calculations for web torque have been based on the assumption that the torque in the web is $l/2R$ times the torque in the journal, l representing the distance between web axes, R representing the distance between the axes of pin and journal. It is believed that this assumption will give approximately correct results in stress calculation. The factor K, listed in Table VI-C when applied as a correction lowers somewhat the calculated value for web torque, but to this corrected value the value of web shear must be added. The total is not far from the results given in Table IV. Even if equivalent web lengths 50 per cent greater had been used in calculating K, the values of K would not have been greatly diminished. It is believed, therefore, that crankshafts should be designed on the assumption that the web torque will be about $l/2R$ times the journal torque.

SUMMARY AND CONCLUSIONS

In a crankshaft with unrestrained journals under uniform journal torque the stresses in journals and pins are purely torsional and the stresses in the webs are purely transverse bending.

In a crankshaft with restrained journals under uniform journal torque restraint of lateral movement of journals may or may not cause torsional stress in the webs. Whether such stress is produced depends on the type of shaft. In a four throw—three bearing shaft, web torque is not thus produced. In a four throw—five bearing shaft or in a six throw—seven bearing shaft, web torque is thus produced.

In a shaft of a type subject to web torque an approximate estimate of web torque may be made by multiplying the value for journal torque by $l/2R$, in which l represents the distance between the axes of the two pins adjacent to a web and R represents the distance between axes of pin and journal.

Study of typical aeroplane crankshaft failures shows that, when the calculated transverse web bending stress is less than the calculated resultant stress in the web due chiefly to torque, fracture starts at a re-entrant angle between a web and either a pin or a journal. Such fractures are not due to transverse bending stresses.

In a few crankshafts of old design the calculated transverse bending stress in the web is greater than the calculated resultant shear stress. Such shafts may fail by transverse bending fracture of a web.

In marine and submarine crankshafts with few exceptions the nominal stress distribution is much more uniform than in aeroplane shafts. Such shafts, if they contain radial oil holes usually fail in a journal, the crack starting in an oil hole. If oil holes are absent they may fail in a web.

In radial oil holes in shafts the stress concentration varies somewhat with the class of steel used. In the class of steel used in aeroplane crankshafts, the stress concentration is about two to one.

It is desirable that crankshafts be so designed that, with web torque calculated as described above the stress distribution will be as uniform as possible.

RUBBER SLEEVES FOR THE PROTECTION OF STEEL SHAFTING FROM CORROSION.

By Commander Paul E. Dampman, U. S. Navy, Member, Engineer Officer, Navy Yard, Boston.

In a vessel with underwater body sheathed with copper, the protection of the exposed portions of her shafting from galvanic corrosion is of the utmost importance. Heretofore, the method of obtaining this protection was by means of composition sleeves, shrunk on to the shaft, in sections of such length as could be conveniently handled. If in one piece, and free from porosity, this forms an ideal protection; however, it is rarely possible to make these in one piece, and efficient means of sealing the joints between the sections of sleeve, so as to make a joint which will be and remain tight, are imperative.

The details of the joint used up to the present time are as shown in Figure 1. The annular ring, between the overlap

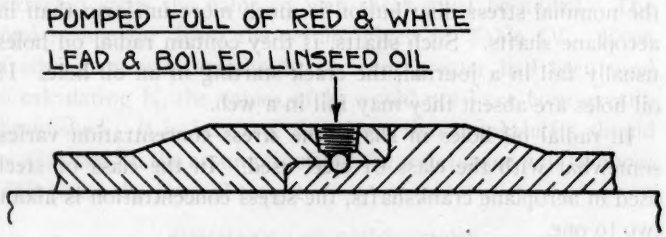


Fig. 1.—Old Method of Packing Sleeves.

of adjoining sections, was pumped full of red and white lead and boiled linseed oil. It will be noted that the thickness of the sleeve is greatest at the joint; the purpose of this was to

permit the twisting of the sleeve, due to torsion, to come between the joints. However, in practice, this did not work out according to theory. With the working of the shaft, alternately ahead and astern, the joint loosened up, the red and white lead hardened and crumbled, salt water obtained admission to the shaft and active corrosion commenced immediately, and in a very short time, the shaft was so weakened as to crack or fracture entirely. Since October, 1922, the Boston Navy Yard has had to replace seven propeller shafts and two stern tube shafts on the *Denver, Tacoma, Cleveland* and *Galveston.*

The *Leviathan* was recently docked for the U. S. Shipping Board in the large dry dock at South Boston. Among other work done on her at this time, one of her propeller shafts was replaced by a spare, on account of a crack in one of the composition sleeves. It was noticed on this shaft that a space of approximately 18 inches between two adjacent composition sleeves was filled in with a rubber sleeve. This had evidently been in place since the vessel was built, or at least seven years, and was in perfect condition except for a cut made by the hoisting slings. It was firmly adherent to the steel beneath, which was in perfect condition with no trace of corrosion.

On account of the cut in this rubber sleeve, it was necessary to replace it. After an examination of the old sleeve, the U. S. Rubber Company declared their ability to replace it and a contract was made and satisfactorily executed by that company.

On docking the U. S. S. *Denver,* in December, 1923, it was noticed that, on her starboard shaft, much rust was issuing from the joints in the composition sleeves on both the propeller and stern tube shafts, indicating that serious corrosion was taking place beneath. Removal of the sleeve confirmed the fears of the Yard, for the stern tube shaft was found pitted to a depth of one-half inch in several places, the pits being from 6 inches to 10 inches in length. (See photograph, Figure 2.) Cracks in the steel were found extending into the shaft at the bottom of these pits. The shaft was in such condition that it was necessary to condemn it. The propeller

shaft, which had been installed new in April, 1923, was found to be pitted circumferentially under each joint in the sleeves, to a depth of approximately one-eighth inch. This shaft, however, was not considered to be so seriously weakened as to prevent its further service, and it was continued in use.

In view of the demonstrated inefficiency of the old type of joint, and having in mind the employment of a rubber sleeve on the *Leviathan's* shaft, the authority of the Bureau of Engineering to employ rubber sleeved joints on the *Denver's* replacement shafts was requested and obtained. A new spare stern tube shaft, completely sleeved as per old specifications, was drawn from store to replace the condemned shaft. On this it was decided to eliminate the old type joints between the sections of sleeve and to substitute therefor a rubber joint of approximately 4 inches in length. On the propeller shaft all of the composition sleeves were removed, except the after strut bearing sleeve, and a short section near the coupling, and it was decided to substitute for the sleeves removed a continuous rubber sleeve approximately 13 feet long.

A contract was entered into with the U. S. Rubber Company to do the work and as it is considered that there are great future possibilities for similar uses of this material, a detailed description of the method of doing the work is given.

The preliminary work consisted of preparing the shafts to receive the rubber composition. The stern tube shaft was 38 feet 2 inches in length and had eight sections of composition sleeve fitted, making seven joints in all. The shaft was centered in a lathe and the sleeves turned off at the joints to a width of four inches, which width was decided upon as the shortest joint which would leave room to properly work the rubber. Each section was underscored at the ends by a 45 degree bevel from outside diameter of sleeve to shaft. The surface of the steel shaft and the undercut of sleeve was rough cut to about 14 threads per inch to aid in the cohesion of the rubber. Figure 3 shows a view of this shaft. In this photo-

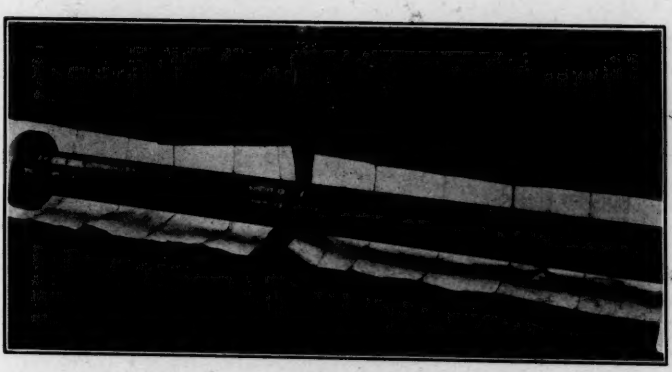


FIG. 2.

FIG. 3.

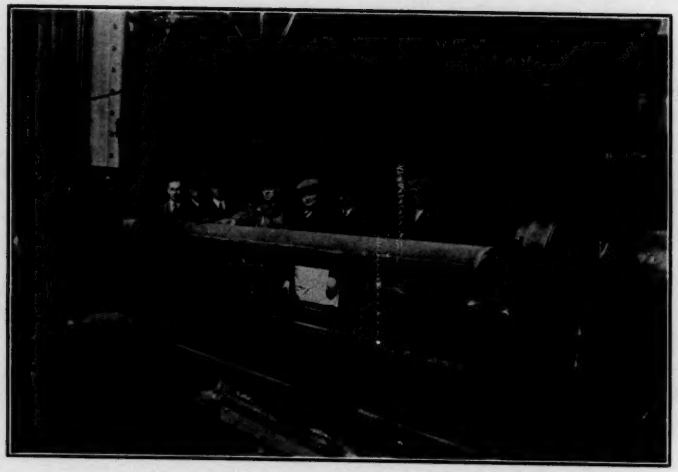


FIG. 4.

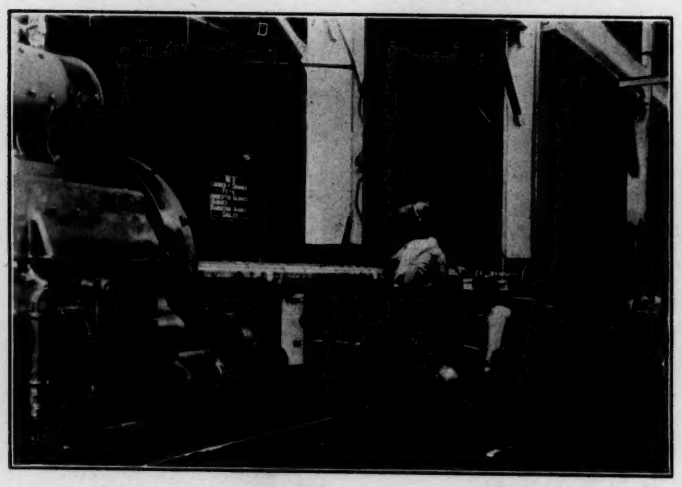


FIG. 5.

graph several joints have already been filled in with rubber, one joint is in process of being filled in, and two at the extreme right have been prepared for the rubber.

In applying the rubber the exposed section of shaft was first well washed with carbon bi-sulphide to remove all oil or grease, and then painted with two coats of rubber cement. The rubber filling, which was supplied in large rolls about a yard wide and one-eighth inch thick, was cut into sheets the proper size to go around the joint, ends scarfed where the joint occurred, and applied one layer at a time with additional strips at the ends to build up a fillet. Each layer as it was applied was well worked with small knurled rollers, great care being taken to get rid of all air bubbles or pockets, the sheet being punctured with awl or knife as necessary. After a layer was in place, it was painted generously with carbon bi-sulphide, as was also the next layer of rubber, which latter was painted on both sides and allowed to stand three minutes before being applied.

Three layers in all were applied, making the thickness of the rubber three-eighths inch, and the whole was then tightly bandaged with strips of muslin wound spirally under considerable tension, until the vulcanization, which proceeded automatically from the chemical action of the carbon bi-sulphide upon the prepared rubber, was completed. This required, at a temperature of 55 degrees F., a period of twenty-four hours. It was stated by the U. S. Rubber Company representative that this time could be much reduced by raising the temperature. At 80 degrees F. twelve hours would be required and at 212 degrees F., only two hours. At the completion of the curing, the muslin bandages were removed and the rubber sleeves carefully inspected.

The long rubber sleeve on the propeller shaft, which measured 12 feet 10 inches in length, was applied in a similar manner. The photograph, Figure 4, shows a view of the rubber sleeve, after it had been applied, and before it had been bandaged.

It was suggested by the representative of the U. S. Rubber Company, who supervised the work, that a layer of rubber impregnated cords such as are used in the manufacture of cord tires, wound tightly around the shaft just before the outside layer of rubber was applied would greatly improve the mechanical strength and durability of the rubber sleeve. It is proposed to incorporate this feature in the next job of a similar nature which develops.

For the long rubber sleeve on the propeller shaft, which was exposed, some form of protection from mechanical injury, and from possible attack by marine growths was considered advisable. (The rubber sleeve on the *Leviathan's* shaft was within the stern tube.) This protection took the form of a sheet brass strip, one-sixteenth inch thick and 3 inches wide, made up in a long coil with joints riveted and brazed, and wound spirally over the rubber sleeve under considerable tension. It was secured to the composition sleeves at each end by means of countersunk head machine screws. The photograph, Figure 5, shows this strip and the method of winding it around the shaft.

The material used consisted of uncured sheet rubber, calendered one-eighth inch thick (U. S. Rubber Company compound G-520), and was furnished in rolls about a yard wide, rolled up with cotton sheeting between the layers. The cement was made by dissolving the above rubber sheet in benzol in proportions of four and one-half pounds of rubber to one gallon of benzol; the benzol being chemically pure and free from carbon bi-sulphide. The carbon bi-sulphide was of the ordinary commercial grade. Precautions should be taken to store the rubber material away from the carbon bi-sulphide as it is affected by the fumes of the latter; also care should be taken in handling both the benzol and bi-sulphide, the vapors of both being explosive when mixed with air.

In addition to the corrosion due to leakage between sleeve joints, there has been considerable difficulty caused by leakage and corrosion at the commencement of the propeller taper. In

order to secure a better and tighter joint at this place, the method shown in the sketch, Figure 6, was developed. This has now been in use for over a year and shafts recently examined have shown no signs of corrosion at the taper.

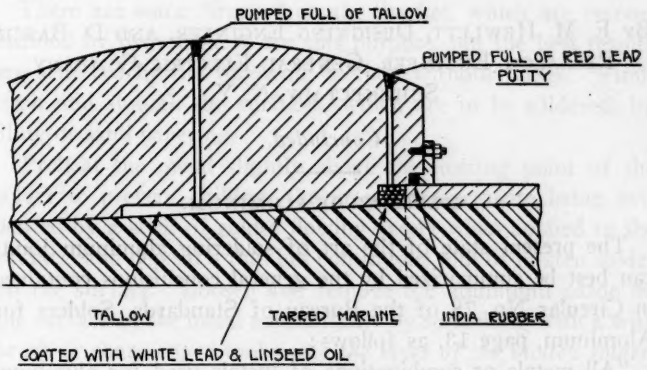


FIG. 6.—METHOD OF PACKING PROPELLER HUB.

NOTE.—Since the above was written, there has been received at the Boston Yard, a spare propeller shaft for the S. S. *George Washington,* which has a rubber sleeve approximately fifteen feet in length. The surface of this sleeve contained numerous fine cracks, such as appear in old rubber, and the sleeve itself appeared hard and brittle. In addition, it was gouged in several places. A new sleeve will be fitted by the Boston Yard, and it is proposed to add a layer of cords to this, so as to produce a cord tire effect.

ALUMINUM AND ALUMINUM ALLOYS FOR USE ON BOARD SHIP.

By E. M. Hewlett, Designing Engineer, and D. Basch, Research Engineer, General Electric Company, Schenectady, N. Y.

Concluded.

SOLDERING OF ALUMINUM.

The present state of the art of soldering aluminum joints can best be summarized by the general conclusions contained in Circular No. 78 of the Bureau of Standards, Solders for Aluminum, page 13, as follows:

"All metals or combinations of metals used for aluminum soldering are electrolytically electropositive to aluminum. A soldered joint is therefore rapidly attacked when exposed to moisture and disintegrated. There is no solder for aluminum of which this is not true. Joints should therefore never be made by soldering unless they are to be protected against corrosion by a paint or varnish, or unless they are quite heavy, such as repairs in castings, where corrosion and disintegration of the exposed surface would be of little consequence. This is especially true for joints exposed to salt atmosphere."

Even if coated with a protective paint or varnish, a soldered joint should never be used where a possible failure of the point would have serious consequences, as no surface protection is one hundred per cent effective and reliable. Aside from these limitations, it is not difficult to produce a reasonably strong joint by soldering, if certain inherently simple precautions are taken. No aluminum solder can bond with the film of oxide, which unfailingly and immediately forms on the surface of aluminum when exposed to air—nor can it bond through the film of oxide with the underlying aluminum.

The whole secret of a good aluminum soldering job is to obtain a clean surface without oxide film and to bring about metallic union between the molten solder and this clean surface before new oxide can form.

There are some "fluxes" on the market, which are recommended by their makers for this purpose, but the best results are achieved by the following method without fluxes. First, clean and smooth the surfaces, which are to be soldered, by filing or sand papering.

Preheat the metal slightly above the melting point of the solder to prevent chilling, the proper temperature being evidenced by a stick of solder flowing freely when applied to the surface, and deposit a thin but continuous film of molten solder on the surface. Loosen and remove the aluminum oxide on the surface of the metal mechanically by scratching with a wire brush or sharp-edged tool *under the layer of the molten solder*, which keeps the air from the aluminum surface during this operation, and thereby prevents the reforming of the oxide, after it has been scraped off. Remove all scum and oxide from the solder with a preheated iron knife or wire and spread the solder evenly over the whole surface.

After having "tinned" the surfaces to be joined in this manner, the actual joint between the tinned surfaces may then be made in the usual manner with a soldering iron and some more solder.

The efficiency of the joint, physical and electro-chemical, depends on the adhesion between the aluminum and the initial layer of solder, and most failures are due to stopping the scraping of the oxide under the molten solder too early.

It should also be noted that a dirty joint is always a weak joint, and care should be taken to remove the scum and oxide from the molten solder.

There is no great trouble in obtaining soldered joints with a tensile strength of about 14,000–15,000 pounds per square inch, tested in a butt joint, if all precautions are preserved and good solders used.

In our own work we found solders of the Zinc-Tin or Zinc-Tin-Aluminum series most reliable, such as 70 per cent Tin, 30 per cent Zinc or 70 per cent Tin, 27 per cent Zinc, 3 per cent Aluminum, although in the mass of commercial aluminum solders, of which we could not test all, there are undoubtedly others of good properties.

The same solders which in butt joints showed a tensile strength of 14,000–15,000 pounds per square inch when new, gave from 10–12,000 pounds after having been exposed to the regular salt spray test for 2 weeks. Other solders tested either failed completely during 2 weeks in salt spray or showed such marked decrease of strength that failure at an early date was to be expected.

The melting point of these solders is around 200 degrees C. so that they can be worked without danger to the aluminum itself.

The Aluminum Company of America is marketing the so-called hard Litot solder for joints exposed to moisture or other corrosive influences, which is used with a special flux to make the solder run into inaccessible parts of an aluminum joint, in the same manner as ordinary solder flows in a well made copper joint. This solder tested in butt joints gave about 12,000 pounds initial tensile strength per square inch with practically no decrease in strength after 2 weeks salt spray test, however, it has a melting point of about 800 degrees F., so that there is considerable danger of burning the aluminum, especially in thin sections, unless great care is taken. To obtain a good joint with hard Litot solder is much more difficult than with the other solders mentioned above.

The danger of gradual failure of soldered joints is, of course, always greatly increased when aluminum is soldered to copper or brass, due to the electrolytic potential difference between the two elements of the joint.

Attempts to plate aluminum with some metal of the same order as brass or copper and then solder brass or copper to the

plated aluminum have so far been unsuccessful, due to galvanic action between the aluminum and the superimposed plate, which is always more or less porous.

WELDING.

Properly welded joints, Aluminum to Aluminum (pure Aluminum or Aluminum Alloys) yield high tensile strength and offer the same resistance to salt water corrosion as the metal in the main body, provided that care is taken to prevent electrolytic potential differences between weld and main body either through differences in chemical composition or in metallic structure.

Welding in its first form was applied to small wires and rods or narrow sheets and consisted of heating the parts to be jointed, at the point of juncture, with a blow-torch to a temperature a little below the fusion point and then hammering or rolling them together until all the oxide was squeezed out. It is still used to a limited extent in the field to join small strands of aluminum wire or cable thin rods, and for similar jobs, but in the main it is superseded by the oxy-acetylene process, which actually fuses the parts to be joined together with the help of a special flux to dissolve the oxide film and to prevent formation of new oxide during the welding operation.

The parts to be joined are first cleaned thoroughly to remove all oil, grease, dirt and loose particles, then they are rough fitted together, for butt joint, lap joint or flange joint, according to the particular nature of the job, whether welding is employed to repair broken parts or to join separate members into one structure according to layout and design, or whether the sections to be joined are heavy or thin. In butt joints the edges of both parts are bevelled off at an angle to one another and brought together, leaving a trough about equal in width to the thickness of the section. The two sections should not actually touch, as the aluminum at the welding heat expands considerably. This method of joining is used most frequently where the thickness of the section is at least $\frac{1}{16}$ inch. In lap

joints the parts to be joined are arranged to overlap a liberal distance, preferably with holes drilled through the upper part or with the edge of the upper part serrated, so as to provide ample space for welding material to get to the lower part throughout the whole width and breadth of the lap. These joints can be made to give extra high strength or to bridge broken out parts.

In flange joints the edges of both parts are bent up at right angles, a height of about three to four times the thickness of the sheet. (This joint is particularly suitable for very thin sections.)

A welding stick, generally and preferably of the same chemical composition as the body material, of convenient length and thickness, is heated and dipped into the flux, which adheres to it in the form of a small tuft. This is melted and made to flow along the rod, so that it gives an even coating about 6 inches long. With this method the chances of getting excess flux in the weld to the detriment of the weld are lessened. The oxy-acetylene flame is then brought to bear on the joint and the parts are fused together while additional metal is fed into the joint from the welding stick, which at the same time supplies the required fluxing action.

There are many commercial fluxes on the market, mostly combinations of alkaline chlorides and fluorides. Just to mention a few out of several which have given good results: one composition contains 40 per cent sodium chloride, 40 per cent potassium chloride, 10 per cent lithium chloride and 10 per cent cryolite, another one 30 per cent sodium chloride, 45 per cent potassium chloride, 15 per cent lithium chloride, 7 per cent potassium fluoride and 3 per cent sodium bisulphate. The heat of the flame should be adjusted to suit the character of the work. On thin sections the oxy-acetylene flame must be handled very carefully to prevent going right through, and lower temperature gas mixtures (oxy coal gas or oxy-hydrogen) are sometimes used.

Especially on long or complicated joints, which are liable to get out of adjustment during the process of welding, it is advisable to spot weld or tack the joint at a few places before making the continuous weld.

After welding, all traces of the flux must be carefully washed off with warm water, or severe corrosion may result.

Care should be taken in welding, that the white, conical part of the flame be kept away from the weld and that the flame be neither oxidizing nor reducing, particularly not oxidizing. Large work should be preheated to prevent contraction and expansion stresses. All work should be suitably supported to counteract any deformation due to fragility of the metal just below its melting point.

There are quite a few tricks in welding, but they are a matter of experience rather than inherent difficulties. A good aluminum welder can be trained in a few hours so that he will produce welds which are just as uniform as castings made from the same metal.

In the case of duralumin or other heat treated alloys, the welds should be heat treated, in order to obtain physical properties commensurate with the strength of the body metal and to prevent corrosion between the heat treated body metal and to untreated weld. Horace C. Knerr, in a report of experimental work done at the Naval Aircraft Factory at Philadelphia, and of results obtained from this work and from welds made by a private firm, states that welds which had not been heat treated were severely attacked by corrosion due to the physical difference between the metal, which had been annealed by the heat of welding and the remaining metal which was in the heat treated state. The weld itself was in no case attacked. Similar specimens, heat treated after welding and subjected to the same corrosion test, were only slightly corroded, having some scattered surface pitting, but none locally along the weld. Specimens were also subjected to a bend test to determine whether the exposure to salt spray had induced brittleness in the welded section. The specimens which

had not been subjected to the corrosion test withstood a 180 degree bend on a diameter equal to about four times their thickness, whereas the non-heat treated corroded specimens withstood only about 135 to 150 degrees bend before cracking. The bending qualities of heat treated welds were not markedly affected by corrosion.

Another method of welding electrically aluminum to aluminum or aluminum to other metals, such as copper, brass, etc., is known as electro-percussive welding. In this process stored up electrical energies are dissipated at extremely rapid and high rates in the form of condenser or reactance coil discharges and applied to the heating of the joint simultaneously with or followed by a percussive forge of the molten surfaces. (See Review of Electro-Percussive Welding, by D. F. Miner in Journal of the A. W. S., July, 1922.) Electro-percussive welding is obviously limited to joints of certain size and shape, but within its range very good and strong joints have been produced.

BURNING-ON.

It is sometimes necessary to make an addition to a casting to complete or to replace a portion, which has been broken off. For this purpose the casting already made is placed in a mold of the proper shape and molten metal poured in. The solid metal at the points to be built up is heated up to a sufficiently high temperature to fuse with the new metal by letting molten metal run through the mold; the outlet hole of the wall is then closed and the mold is allowed to fill up.

Such joints, if carefully made, are equally as strong as the main casting and have the same resistance to salt air corrosion as the material of the main body.

ALUMINUM IN CONTACT WITH OTHER METALS.

Frequently aluminum or aluminum alloys are brought in physical contact with other metals and the question arises to what extent such joints are subject to corrosion when exposed

to salt air. Corrosion cannot take place in the absence of moisture or air, and airtight joints of two dissimilar metals can therefore at the start corrode only at the edges where they are exposed to air and moisture, but such corrosion gradually works its way in and finally destroys the whole joint.

The rate of corrosion between two dissimilar metals, as distinguished from the inherent corrosion of each individual metal, can roughly be expressed as a function of the electrolytic potential difference between the two metals in contact; the metal which is electro positive to the other one being attacked by the galvanic corrosion.

The following table, taken from Ledebur and Bauer (Alloys for commercial applications), gives the potentials of various metals and alloys against the standard calomel electrode in 1 per cent table salt solution at 18 degrees C. after 120 hours:

Magnesium	+	1.598
Magnesium base alloys		
(with 7.3 Zn.)	+	1.516
(with 5.12 Al.)	+	1.480
Zinc	+	1.037
Cast Iron	+	0.760
Wrought Iron	+	0.755
Steel, low carbon	+	0.744
25 per cent Nickel Steel	+	0.581
Cadmium	+	0.741
Aluminum, pure	+	0.737
Aluminum base alloys		
with 4 per cent Cu., 0.5 Mg.	+	0.763
with 4 per cent Cu.	+	0.744
Duralumin	+	0.577
Tin	+	0.422
Copper (pure)	+	0.223
Copper base alloys		
with 10 per cent Sn.	+	0.146
with 26 per cent Zn.	+	0.243
Nickel	+	0.080

We ran extensive tests to check this table by exposing specially prepared contact couples of various commercial metals and alloys to the action of salt spray for 8 weeks, after which time all corrosion products were carefully removed and the loss of weight measured as an indication of the rate of corrosion. The actual test results were found to agree in the main very well with the conclusions deducible from the table. For instance, cast iron specimens were tested in contact with zinc, aluminum and soft copper. Zinc is markedly electro positive to cast iron, cast iron is slightly electro positive to aluminum and more so to copper. The area of all specimens were alike.

The loss in milligrams was as follows, measured in each instance on the cast iron,

When in contact with zinc, 150.
When in contact with cast aluminum, 750.
When in contact with soft copper, 1,850.

In another series of contact couples, comprising hot dip galvanized iron in contact with zinc, cast aluminum and soft copper, the following results were found (measured on the galvanized iron samples).

Loss in milligrams,

When in contact with zinc, 90.
When in contact with cast aluminum, 200.
When in contact with soft copper, 425.

In still another series, comprising cast aluminum in contact with cast iron, zinc, hot galvanized iron and soft copper the results were as follows (measured on the cast aluminum sample).

Loss in milligrams,

When in contact with zinc, 30.
When in contact with hot galvanized iron, 80.
When in contact with cast iron, 100.
When in contact with soft copper, 300.

Obviously it is impossible to determine quantitatively from the table what the corrosion of any metal will be as the result of being joined to another metal—too many other factors enter in besides the electrolytic potential difference. However, it would seem justifiable to gauge the rate of corrosion of any one particular metal or alloy, when in contact with various other metals, relatively from the table.

The figures quoted above from our tests show that there is a vast difference in the rate of corrosion of the same metal when brought in contact with various other metals.

As a matter of construction principle, the coupling of metals widely different in electrolytic potential should be avoided wherever possible. Where it cannot be done, the joints should be protected carefully with paint or varnish to exclude air and moisture, and this protective coating should be renewed at frequent intervals.

ALUMINUM FOR ELECTRICAL PURPOSES.

Aluminum for bus and connection bars, etc., on switchboards has been used in England to a much greater extent than in this country, and they have actual life service data extending over more than 10 years. Aluminum bus bar installations have been in operation in various localities in England, some of them with distinct salt atmosphere, long enough and with such uniformly good results that this fact alone would justify the flat statement that aluminum can safely be used for bus and connection bars on board ship, and that aluminum to aluminum joints, if properly made, have well over 100 per cent efficiency (meaning that the total resistance of the length comprising the joint is less than that of an equal length of the straight conductor) and will maintain it under the conditions prevailing on board ship practically indefinitely. A great deal of work has been done by the British Aluminum Company and the National Physical Laboratory of England (particularly Messrs. S. W. Melsom and H. C. Booth) to determine the best and most reliable method for making aluminum joints, and the

results of their work can be summarized as follows: Only high purity aluminum should be employed as impurities or alloys lower the conductivity and have been responsible for many early troubles with defective joints and corrosion. The contact surfaces must be quite flat; it is advisable to file the surfaces carefully, preferably using a rough cut file and arranging that the file marks on the two bars are at right angles when they are put together. This assures a more thorough bedding of the surfaces and consequently a better contact. Next the two surfaces should be covered with a thick layer of vaseline and cleaned with emery cloth under the grease. The reason for this is that aluminum oxidizes upon the surface with extreme rapidity and a freshly filed surface in a few seconds becomes covered with an invisible layer of oxide which though microscopically thin has an appreciable effect upon the contact resistance. If the surface is cleaned under vaseline, however, the oxide is removed and the vaseline protects the surface from reoxidization from the air. The two bars are then pressed together and the tightening of the bolts squeezes out the vaseline from the points where actual metallic contact is made.

The average lap required to give 100 per cent efficiency depends on the thickness of the bar, but not on the width, as follows:

Thickness of bar	Length of overlap (minimum)
3/16 inch	1.9 inches
1/4 inch	2.2 inches
3/8 inch	2.7 inches

The joint pressure should be at least about 2,000 pounds per square inch. The joint may be made either with clamps or with bolts.

Clamps should be made of high strength and stiff aluminum alloy, and they should be so designed that they do not buckle at the center under the strain at the bolt holes, thereby releasing the pressure at the center of the bar. Bolts should preferably

be made of heat treated Duralumin or equivalent metal. In general, bolted joints will be more effective than clamped joints. Large washers should be used with bolts at both sides of the bus. In determining the size of the aluminum conductor to carry the required current, notice should be taken of the fact that the conductivity of pure aluminum is about 60 per cent of copper. The cross section of aluminum should be 1.66 times that of copper in order to carry the same current with the same voltage drop. However, if the desired increase in area is taken care of by using bars of greater width, but the same thickness as the copper, the increased radiation of the aluminum bar will permit an appreciable saving of actual cross section without raising the temperature. In laminated joints, of course, the increase in useful radiation will be less than with single bars.

Aluminum to aluminum joints made in the manner described above do not need to be protected in any way against salt air corrosion. Sometimes aluminum tubing or rods are used as electrical conductors. They should either have aluminum terminals welded to their ends or should be flattened at the ends and then joined largely in the same manner as described for bars.

When it becomes necessary to join aluminum to copper, the joint should always be carefully protected.

Aluminum should not be used for any but permanently closed joints. Switch or circuit breaker contacts, clip contacts, threaded stud and nut joints do not lend themselves to the employment of aluminum, due to the fact that the aluminum is bound to oxidize on the contact surface when the contact is opened and to increase the contact resistance to a point where heating will become very severe.

One important field of application of aluminum in electrical apparatus is based on the fact that aluminum and aluminum alloys are non-magnetic, same as copper and the brasses. Electrical apparatus for alternating current, especially when the magnitude of the current handled by the apparatus is high,

should preferably not contain any steel, iron or ferrous alloys, in order to prevent inductive heating. In the past, copper, brasses and bronzes have been used for structural parts (not directly carrying the main current) in alternating current apparatus for heavy currents, but the tendency is now towards the employment of suitable aluminum alloys, since they are cheaper and lighter than the cuprous alloys. Of course there are still quite a few places in this field where aluminum cannot be used—for instance, where the part is exposed to the electric arc—where threaded joints are operated very frequently under heavy strain, or where hard, unyielding surfaces are required.

The conductivity of the commercial cast aluminum alloys is between 30 and 35 per cent of copper and, therefore, quite a bit higher than that of the ordinary brasses and bronzes.

CONCLUSION.

The use of aluminum on board ship (naval or mercantile) is in its infancy. Its vast possibilities in the reduction of weight, its attractive non-corrosive properties, its adaptability to simple and improved foundry processes, which cannot be applied to the ferrous or cuprous alloys, are just beginning to be appreciated by the naval designer.

A big field of metallurgical research and technological development is still to be covered, however, before aluminum and its alloys will be fully established in their rightful place.

Alloys of greater stiffness and tensile strength, the fabrication of structural shapes and sheets of large enough dimensions to match what can be obtained in steel and iron, and the elimination of the variables in foundry and rolling mill which make for wide range of properties in the same material are some of the most important problems still before us.

The influence of the navy on the consummation of this object can and undoubtedly will be very great. The navy has always been more exacting and careful in its specifications and requirements for its material than any other large consumer. It has always stood out against "run of mine" production, and

has fathered and successfully insisted on methods of fabrication and manufacture which the producer thought were too far ahead of the demands of the industry.

The late Andrew Carnegie said that the American Steel Industry was built up on the United States Navy, whose contract specifications and inspection work made steel what it is today.

We hope and trust that the U. S. Navy will do the same for the Aluminum Industry.

NOTE: In the first instalment of this article published in the November, 1923, JOURNAL on page 682, 9th line from the bottom, 1½ per cent should read 3 per cent.

DISCUSSION.

By Captain P. B. Dungan, U. S. N., Member, Officer in Charge Engineering Experiment Station. Annapolis, Md.

I have read with interest the article entitled "Method of Balancing Turbine Rotors of the Scout Cruiser *Raleigh*," by Commander Carlos Bean in the February issue of the Journal.

The apparatus described for the determination of plane and amount of unbalance is most ingenious and should result in materially reducing the time required for the determination of this data; as it evidently lends itself to the alteration of the plane in which the correcting weights are applied, as well as the variation of the amount of correction without stopping the spinning of the apparatus being balanced.

In the case of a large turbine rotor, this starting and stopping constitutes a very large percentage of the time required.

It is not understood, however, why it should be necessary to devote any time whatever to the elaborate routine described in correcting the static unbalance by means of the old and—in the case of machinery of this size—inaccurate method of knife edges.

The author's distinction between static and dynamic unbalance made in the opening paragraph is not accurate. The vibratory effect produced by static unbalance is entirely different from that produced by dynamic unbalance. An object may be in unbalance either statically or dynamically or both, but dynamic unbalance cannot be considered as the result of a "residual" static unbalance. A body may be in perfect static balance, yet have excessive dynamic unbalance.

The static balance may be made perfect, but if the static correction weights are not very accurately and correctly placed as regards their position relative to the center of gravity, these weights may introduce additional and serious dynamic unbalance.

A very clear and complete discussion of what constitutes static and dynamic unbalance is contained in an article on balancing apparatus in the JOURNAL of May, 1919, by F. G. Hechler, Mechanical Engineer.

In the procedure as outlined in this article, the static unbalance is partially corrected on knife edges—the amount of dynamic unbalance being possibly increased or diminished by this process—then by the second operation the dynamic unbalance and the remaining portion of the static unbalance is compensated by weights whose amount and position are determined by the spinning method.

This duplication of operations is, in this case, not necessary and there is no reason why the first operation on knife edges should not be omitted entirely, and the entire job simplified and shortened by doing all balancing by spinning with the correcting weights. This procedure would produce accurate balance, in this case, where the correcting weights are applied at each end of the rotor simultaneously.

It is considered that the article would have been of more practical benefit to anyone endeavoring to follow this method, if a more complete description had been given of the procedure in transferring these compensating weights to the turbine rotor itself. The compensating force created by these weights is due to the moment of the couple acting about the center of gravity and in transferring this compensating effect to the rotor itself, the weights must be so proportioned and placed as to produce an equal moment in their new locations.

It may be interesting to note that an accurate indication as to the amount of static unbalance corrected by the compensating weights in spinning is given by the angle formed between the two end clamps.

If these are in the same longitudinal plane but 180 degrees apart, the correction is entirely dynamic; if their angular displacement is less than 180 degrees, there is a portion of static unbalance being compensated—the proportional amount being greater the more the angular displacement between the weights differs from 180 degrees.

If the weights should be in the same plane and same direction, the correction is entirely static. It may be remarked that a condition of entirely dynamic or entirely static will practically never occur—and in pieces of apparatus of the size of the rotors for one of the Scout Cruisers, it will invariably be found that a very appreciable amount of static unbalance remains for compensation, regardless of the most careful balancing on knife edges beforehand.

REPLY.

By COMMANDER CARLOS BEAN, U. S. N., MEMBER.
INSPECTOR OF MACHINERY, QUINCY, MASS.

Referring to Captain Dungan's discussion of my article on the subject of balancing statically and dynamically, the turbine rotors of the *Raleigh*, I believe, from experiments that were made and most carefully observed by the best talent of the contractor's and the Inspector of Machinery's office, that it is most essential to balance each rotor statically by the method described. I fully agree with Captain Dungan that the old method of static balancing on 1 inch or wider knife edges is useless. Turbine rotors that were given a static balance on these 1 inch knife edges invariably showed dynamic unbalance, and it was necessary as the result of the spinning test to apply weights on the turbine wheels at both ends of the rotor. To avoid the time consumed and trouble in finding the amount and location of the weights, and reduce the number, the refined method of static balancing as described in my article was suggested and tried out.

The results were most encouraging. As stated, two of the I. P. rotors showed perfect dynamic balance. The four H. P. rotors and the two other I.P. rotors required, as the result of the spinning test, about six ounces each to place them in perfect dynamic balance, and in each case, it was only necessary to apply one weight. It is a decided advantage in applying this weight to keep it down to a minimum.

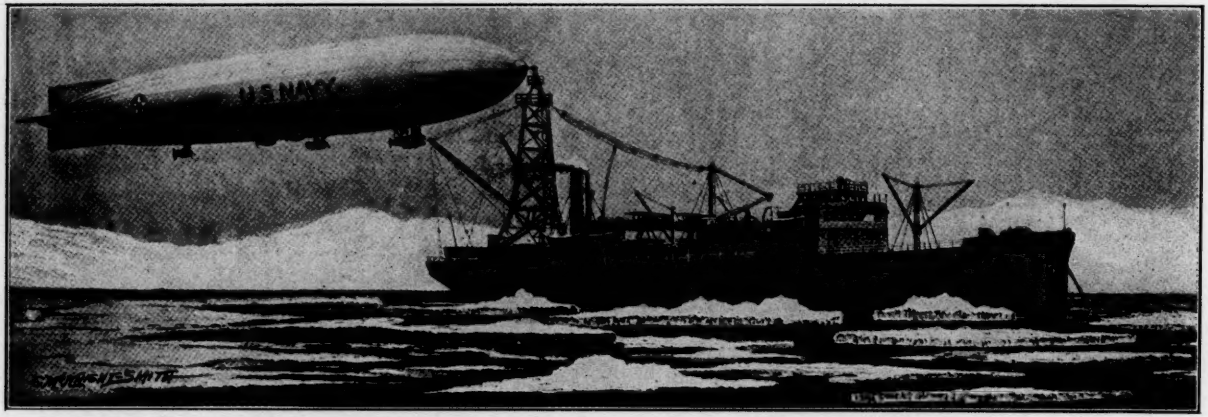
Further, as an experiment, one of the H.P. rotors was given a spinning test without having been balanced statically. It showed marked vibration at both ends. It was removed from its casing and balanced statically. When it was again given a

spinning test, the vibration was greatly reduced, and the application of a six ounce weight at one end of the rotor removed all dynamic unbalance. In other words, a rotor which is out of balance statically is undeniably out of balance dynamically. Why not remove the static unbalance and thereby facilitate removal of the dynamic unbalance. "An ounce of prevention is worth a pound of cure." The method of manufacture of these Curtis Units is such that any small amount of unbalance can be confidently corrected statically, and it has never been found necessary to make counter-corrections developed on the spinning test.

As to placing the weights on the rotor, the angular position of the balancing arm indicates the angular location of the weight. The hand indicator and elongated arms in contact with the casing give the approximate location of points of greatest vibration. If the weight necessary is over six ounces, split it up and place parts on adjacent wheels. The rotor is then given another spinning test for balance.

The design of the Curtis turbine, having equalizing holes in the wheels, is particularly well adapted to allow securing the weight easily and on any wheel temporarily for trial and verification before being finally secured for full due. The above scheme was carried out in the *Raleigh's* turbines. The complete operation of static and dynamic balancing takes from two to five days.

It is interesting to note that the main engines of the *Raleigh* on her preliminary acceptance trials were remarkably free from vibration.



U. S. Navy Dirigible "Shenandoah" Swinging at the Tall and Sturdily Constructed Mooring Mast of a 450-foot Supply Ship.

NOTES.

THE MOORING MAST IS THE THING.

By J. Bernard Walker.

Once launched in the air, the airship should stay there, returning to its hangar only when major repairs or general overhauling and repainting are necessary. The hangar should be regarded as having the same relation to the airship as the drydock has to the ship of the sea. Foremost among the perils to which a seagoing ship is exposed is that very *terra firma* upon which she was built. If her great fabric so much as touch the coast or the sea bottom, it is liable to be badly broken or wrecked beyond repair. When she enters port she makes fast to a mooring, where she may swing head to tide or wind; or she is moved slowly and gently to her pier, and protected from rude contact with its masonry by carefully disposed fenders.

If mother earth is dangerous to the seaship, it is doubly so to the airship. If contact with the unyielding ground is perilous to the frail shell of the ship, it is immeasurably more perilous to the delicate shell of a dirigible; as the loss of many of the early Zeppelins so clearly proved. More than one of these was destroyed as it was entering its shed—a puff of wind serving to swing the hull against the doorway and break its back. Others were lost when they came to earth, during a trip, for repairs or replenishment of fuel.

These disasters, and they were frequent, proved that moorings were as necessary for the ship of the air as they were for the ship of the sea, and that it was as foolish to ground an airship upon the land at the end of a voyage as it would be for a skipper to run his ship ashore every time he entered port.

As long ago as 1910 the writer drew attention in the "Scientific American" to these self-evident facts and suggested that the only rational way to handle the airship, between cruises, was to moor it to a rigid and lofty mast. The suggestion was first put into practice several years later by the Vickers, Maxim firm, and its practicability was proved when airships in England remained moored in the open for several weeks on end, and rode out gales in which the wind rose at times to 50 and even 60 miles an hour.

The fact that a 70-mile gust tore out the nose-cap of *Shenandoah* proves nothing against the mooring mast. What it does prove is that we must not apply a heavy concentration of stress at a single point in the delicate shell of an airship, unless provision is made for the immediate distribution of that stress throughout a large contiguous area of the hull. It was, in part, the omission to do this that caused the failure. The nose-cap is a cone of metal plate, which stood up to its work. To this were riveted the converging ends of the relatively light longitudinal girders. These tore apart just beyond the nose-cap. If there was to be a break it was certain to come just where it did.

Those of us who rode the bicycle in the early days of its development will remember how frequently the diamond frame failed, by bending or fracture, an inch or so back of the steering post. Connection between frame and post was made by inserting a short stub-end of tubing in the frame and brazing the whole together. The fracture always came at the point where the stub finished. The *sudden* decrease in section was an invitation to fracture. The end of the stub was tapered into four fingers of diminishing width. There was no longer a sudden reduction to section; the resistance to bending stress was distributed further along the tube; and breakages ceased.

Now like treatment would produce like results in this mooring problem. Distribution of stress could be secured by leading light cables or wires of plough steel from the nose-cap along and within the longitudinal girders and attaching them to the girders at the successive circular frames. Four cables should lead to the first ring, four to the next, and so on until the sixth ring was reached. By spacing the points of attachment at 90 degrees around each ring, the pull of the mooring cable would be distributed, *evenly*, over the first 75 feet of the ship.

To ensure immediate and equal transmission of stress, the cables, before insertion, should be stretched to just beyond the elastic limit and given a permanent set. Each should have a turnbuckle at its mid-length, and all of the cables, or wires, should be set up to the same tension. The tearing loose of *Shenandoah* due to a sudden dynamic stress suggests that we may go to the sea for another suggestion. To ease a sudden strain in the towing cables, powerful tugs make use of towing winches. These allow the cable to pay out when the pull exceeds a certain limit, taking it up when the strain eases up. The winches at the base of the mooring mast should embody this feature. This flexibility, coupled with a distribution of the cable stress over a large area of the ship, as suggested above, would afford reasonable confidence that *Shenandoah* could ride out any gale that might strike her during her Arctic venture.

We present a picture of the mooring ship which will be sent north ahead of the *Shenandoah*, whenever she makes her trip. A mooring mast carried on a vessel would be liable to swing considerably, even if the ship were in sheltered waters; for the effect of a sea reaches a long distance up any harbor or inlet. Consequently, it will be inexpedient to moor *Shenandoah* closely against the mooring mast, except for such brief periods of time as it may be necessary to do so. The movement at the top of a mast 175 to 200 feet above the water is considerable, even in a gentle roll.

The bridge of a destroyer rolling ninety degrees or more sweeps through the air, at the top of the roll, at a speed of over twenty-five miles an hour. Her bridge, however, is only about 25 feet above the sea. The top of the mooring mast will be, say, 175 feet above the water. The ship's period of roll would be slower; but the greater arc swept through would go far to compensate for this. If *Shenandoah* were riding transversely to the axis of the ship she would be alternately pulled forward and thrust back at a speed which, due to her weight of some 50 tons, would set up heavy inertia stresses which she was never designed to meet. If she were riding parallel to the rolling ship, she would be so heavily strained, transversely, that her girders might be buckled and the ship wrecked.

The officers in charge of the expedition favor the plan of letting the *Shenandoah*, after she has been refueled, ride on several hundred feet of cable. The sag of the cable would act with a cushioning effect in heavy gusts of wind, and any rolling movement of the masthead would be damped out. Moreover, the *Shenandoah*, riding high, would be in less danger of contact with the water or with the ship itself.—"Scientific American," April, 1924.

THE COMPENSATION IN WEIGHT OF FUEL CONSUMPTION OF AIRSHIPS.

By Col. G. A. Crocco, Rome, Italy.

In a previous article we have pointed out that the losses of gas due to the maneuver of a dirigible are in the order of magnitude of one hundred times the corresponding losses due to the permeability of the gas bag and we have emphasized the point that either in helium filled or hydrogen filled airships these losses must be eliminated if commercial operation of airships is to become a practical possibility.

It is obvious that these losses are of a great importance only in the case of aerial navigation over long distances. As long as the weight of fuel and oil lost during a trip is only a limited fraction of the total lifting force of the ship it is always possible to compensate such losses with the action of the rudder. When, however, a continuous flight is made over a long distance without any intermediate stop, the loss of weight of the fuel used up during the trip becomes a large fraction of the lifting force and the dynamic sustentation is not possible any more. We must therefore provide for such losses in another way. Two means are most suitable for the purpose: *the condensation of combustion water of the motors* and the *thermic sustentation.* In the present article we will consider the first and in another article we will take up the investigation of the second method.

CONDENSATION OF COMBUSTION WATER.

The fuel used by airships contains a large percentage of hydrogen and therefore a considerable amount of water is produced during the combustion. An average of 150 grams of hydrogen are contained in 1,000 grams of fuel. These will produce 1,350 grams of water vapors and theoretically it is possible to condensate at least 1,000 grams of water, thus compensating the loss of weight of the fuel used up. The difficulties inherent to such undertaking are dependent upon the temperature and the humidity of the surrounding air and become more prominent with dry and warm air and less so in damp and cold air. It would be particularly easy to condensate 1,000 grams of water if the air admitted to the motors is already saturated with water, no matter what is the temperature. In practice, however, we cannot depend upon having always such a desirable state of affairs and the method suggested might appear to be only a partial solution of our problem, unless we consider beforehand the theory involved.

Let us call P the weight of air needed for the combustion of 1,000 grams of fuel at the initial temperature t_o the humidity C_o and the atmospheric pressure H_o. Calling F_o the tension of saturation corresponding to the temperature t_o, the volume of air corresponding to the weight P is given by:

$$V = \frac{P\,(1 + \alpha t_o)}{1.7\,(H_o - .377\,C_o\,F_o)}$$

and the weight ρ_o of the water vapors contained in this volume of air will be given by:

$$\rho_o = 1.06\,\frac{C_o\,F_o}{1 + \alpha t_o}\,V = \frac{.623\,P}{\dfrac{H_o}{C_o\,F_o} - .377}$$

or approximately:

$$\rho_o = .623\,P\,\frac{C_o\,F_o}{H_o}; \quad \ldots\ldots\ (1)$$

To this weight must be added 1,350 grams of water vapors developed during the combustion. If we cool the exhaust gas to a temperature t and a pressure H, the gas will be exhausted at a tension F of saturation and will contain a weight of water vapors of about .623 P $\frac{F}{H}$ and the weight of water condensated will be: $\rho_0 = 1.350 - .623$

$$\frac{PF}{H} - \frac{P_o C_o F_o}{H_o}; \ldots\ldots (2)$$

For every practical purpose we can put $P = P_o$ in the above formula and, therefore, in order to have a weight $\rho = 1{,}000$ grams of water condensated we must have:

$$\frac{F}{H} = \frac{C_o F_o}{H_o} + \frac{.56}{P_o} \ldots\ldots (3)$$

From the above formula we can derive the enunciation of the following theoretical principle:

"No matter what is the initial temperature and degree of humidity of the air, a relation exists between the pressure and the tension of saturation of exhaust gases which allows the condensation of a weight of water equal to the weight of fuel used up."

We must now prove that the relation expressed by formula 3 is practically possible within ordinary meteorological limits.

For a complete analysis of this problem the writer would refer the readers of "International Aeronautics" to the paper presented by the writer to the National Academy of Lincei (Vol. XXXII, bulletin No. 8, February 4th, 1923). The conclusions arrived at in this paper are based upon the investigations made by the Meteorological Observatory of Rome all over the world. These observations have proved that very seldom temperatures of more than 30 degrees and a degree of humidity below 50 per cent are found. Over the European and the American continents an average of 60 per cent of humidity of the air is to be found. We must also consider that as the altitude increases the absolute degree of humidity and the temperature of the air also decrease in such a way as to compensate each other. Higher temperatures are to be found only for a fraction of the day and year and therefore results of calculation applying to exceptionally unfavorable conditions become negligible in the operation of dirigibles all the year around, which is the condition required for the successful operation of commercial airships.

For these reasons we can consider as a practical possibility the adoption of apparatus allowing in *every case* to recuperate the total weight of fuel used up in aerial navigation.

In the paper mentioned above the writer has considered two methods for reducing the power needed for the condensation of the water content of exhaust gases: One is based upon the throttling of exhaust gases at the outlet, thus forming a back pressure in the motor and the other is based upon the compression of exhaust gases with a separate compressor.

The first method is the simplest of the two because it does not require the addition of a mechanical apparatus; on the other hand, however, the power absorbed is greater than in the case of the second method.

The second method offers the advantage of allowing the compensation between the work of compression and expansion which resolves itself into a saving of power. This compensation can be obtained by coupling on the same motor shaft a turbo-compressor and a turbo-motor. The turbo-compressor takes in the exhaust gases after they have been cooled off and, after compressing them, exhausts them into the turbo-motor after cooling.

If we admit a 60 per cent efficiency of both machines we find that the *practical* work of compression is reduced to the limits of the *theoretical* work required.

The powers required for the surcompression are as follows:

Surcompression in m. of water	Percentage of excess power required with Motor back-pressure method	Separate compression method
1	2.87	1.5
2	5.75	3
3	8.60	4.3
4	11.90	5.5

We wish to emphasize once more that the high values of excess power are only required for short trips and for a limited number of days of operation during one year. For an all year around service the *annual per cent* excess power required for the condensation of a weight of water equal to the weight of fuel burned up during navigation is in the average of the order of magnitude of one per cent only.

For this reason, it is probably to be preferred (except in the case of particularly warm and dry climates), the motor back pressure method which avoids all mechanical complications.—"International Aeronautics," Dec. '23–Jan. '24.

THE COURSE TRACER AND AUTOMATIC LOG OF LIEUT. BAULE.

The above is the title of an article published in "The Yacht" in reply to numerous letters asking for details of this apparatus, as used on the ship *Baule*.

The following is a description of the two machines. The tracer of courses consists of a map, mounted securely on a table upon which can be made to transcribe the course set for the ship, by means of a moving track and connections as shown, or as desired.

Figure 1 is a sectional view of the Nautograph. The table (1) made of soft iron, is secured by supports (2), and carries the map (3) in an invariable position. In the center of the apparatus is a column (4), upon which sets a cast tray (5), surmounted by a soft iron plate (6), which is provided with rollers (7) and (8) and a groove or track for guiding its motion. Below the table is an arm (9) to which is attached a spring (10) secured to a lug (11) of the tray. Connected to this arm is a metal rod (12) for the electro magnet (13). This electro magnet is mounted on the base of the tray (5), and can be adjusted by the hand wheel and screw (14) and (15) respectively. The column (4) cast with a gear wheel (16) meshes in a worm (17), and derives its motion from an electric motor (18), the motor being operated by the ship's compass. The connection between the worm and motor comprises another invention of Mr. Baule's, but may be operated by any other system. An electric magnetic check for the gear wheel is represented by a pawl (19), held in place by a spring (20). This pawl is operated by an electric magnet (21) and made to engage with the teeth of the gear wheel (16). The movable tracer is operated by an electro magnet rod of soft iron (22) and two coils (23) and (24), in a sense inverted.

In the earlier built tracers, a bent arm operated by this electro travelled along the support (2), and a stylo placed at the end traced the course on a map. Mr. Baule has simplified this by the use of a metal ball (25),

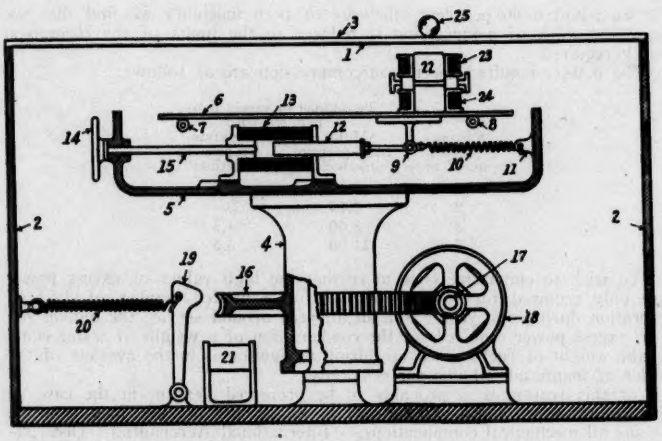


Fig I.

which is coated with ink or made black with smoke. The contact of the electro (22), bearing continually beneath it, and moving along with the electro, indicates the position of the ship on the map.

The operation of the apparatus is as follows:—When the ship is set on a determined course, the motor (18) being started by the compass, motion is imparted to the column (4), causing the sliding of the plate (6) in a direction which produces an impress on the map (3), following the course of the ship. The electro magnet (13) operates at regular intervals, either by a watch or a log, and causes the table to advance to the left a distance corresponding to the stroke of the rod (12) and the position of the coil (13), this position having been regulated previously by the hand wheel (14) in such a manner that the course corresponds on the map, a distance run by the ship during an interval of time of two successive impulses. During the time the table (6) is moving to the left, the electro magnet (22) is excited by the coil (24), making contact with the table; the ball (25) then traces on the map the course the ship is running. During the rest of the time the electro magnet (22), excited by the coil (23), is in contact with the table of the map. When the electro is no longer excited, the spring (10) contracts, thereby moving the table (6) back to its former position to the right. A commutator not shown, but easily imagined, can be operated by the rod (12) of the electro magnet (13), and excited at any moment desired by the coils (23) and (24).

The worm (17) is provided with an elastic connection with the shaft of the motor (18) in such a way as to avoid any continual variations of the compass translated by the intermittent movements of the table (6). The gear wheel (16) has a regular turning interval, corresponding to the changes of the course, which is the mean unit period of time. At this period the pawl (19) and magnet (21) arrests the gear wheel (16) regularly. During this time the worm (17) is at rest, but the elastic transmission between the worm and the motor (18) effectually takes up all the changes during the run.

Before the table (6) is in motion, the pawl (19) is at rest, and by means of the elastic transmission with the motor, the worm (17) transmits at one turn of the gear wheel (16), the changes of courses that would have been made during the period of time unit preceding.

The apparatus for tracing courses can be located at different points of a ship for plotting on a map the route travelled.

Mr. Baule makes the principal parts jointly, of a stem passing through a socket; that stem is provided with a tooth rack to work with a pinion and a shaft parallel to that of the socket. The socket shaft and pinion is dependent then on the changes in movement that suffices to transmit, by means of the electric motor and the fittings placed next to the table of the receiving or repeating posts.

Granting that the same principle, as applied on the *Baule*, is used in tracing the course followed by another ship other than the one carrying the apparatus, it gives the following relative changes of the two ships and solves the continued problem of the direction of propulsion. In fact the Tracer of Courses is a complete apparatus, conceived in 1916, and has been tried on different kinds of ships, and is now adopted in the French Navy.

Baule's early metal log system consisted in dropping in the wake of a ship an inert float with which one measured distance.

The ship's log is based on the time taken by an air bubble to travel from one end of a tube to a contact stud. The measure of time is effected electrically. The moment the bubble is in advance of the tube, a chronograph is automatically put in motion. When the bubble passes the contact stud, it cuts the current and insulates the contact stud instantly. That cut out is used as a means for stopping automatically the chronograph, then we obtain the data.

In dividing the known travel of the bubble by the time taken, we find the mean speed of the bubble, which is the mean speed of the ship, using a coefficient taken in general for a determined speed. This coefficient is called the tarage of the log.

Figure 2 is composed of a rectangular brass tube of 20 meters, mounted on vertical irons which maintain an immersion of 0 meters to 70.

The tube "A" is 36 millimeters inside diameter and 2 millimeters thick, and is open at both ends. The air bubble is destroyed by a blast of air from pipe "B." A few centimeters from the other end of the tube is located a small well "C" of which the upper end is above the water. In this well is

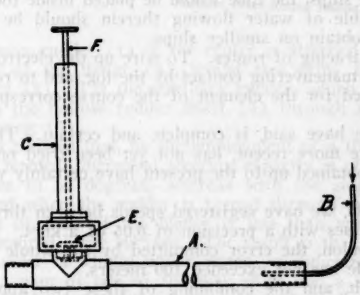


Fig II

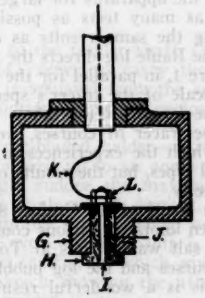


Fig III

a rod "F," at one end of which is the contact stud "E." The rod "F" is threaded to regulate the distance of the contact stud from the inside of the tube "A."

Figure 3 is a section of the contact stud "E," consisting of a metal box "G." A porcelain tube "H" insulates the part conducting the contact stud, which is formed of platinum wire "I," which wire is forced into the brass screw "J." A conductor "K" is secured by nuts "L," permitting communication with the movable table of the apparatus. The maneuvering table governs the flow of air and its measure. The air is supplied from a tank having a capacity of one or two liters, tested to 60 kilograms, connected with a compressor functioning to about 15 kilograms.

The circuit is closed by the contact stud "I" and the sea normally; it is cut off each time a bubble passes and instantly insulates the contact stud. Electro systems and relays automatically produce another bubble at the end "B" each time a bubble passes under the contact stud "I."

The time constant of the apparatus is the time taken for the bubble to pass from end "B" to the contact stud. We can consider the formation of a new bubble at the instant the preceding one reaches the contact stud, owing to the rapidity of electrical connections, the infinitesimal time taken for the new bubble to form can be neglected.

To find the ship's speed if the log has been gauged, it suffices to know the interval of time between two bubbles, or the mean interval of a great number of bubbles. On the other hand, the number of bubbles is proportional to the distance run of the ship. In short, knowing this number gives a constant very close to the total effective run during the time the log functioned.

The disposition of the measure—we will not go into this detail because it would lengthen the article too much—however, we have noted

First—To estimate the interval between two consecutive bubbles.

Second—To take the mean of the greatest number of intervals.

Third—For the total number of runs.

The log can naturally transmit these indications, and as many as one desires from any station on the ship.

Perfected details have stopped parasitic action of the bubbles when the air is flowing. This trouble had often caused several bubbles to pass on the first count only. Orifices at the rear of the tube suppressed the variation of pressure due to the plate satellites, which blurred the indications of the log.

In the apparatus for large ships, the tube would be placed inside the shell, and as many tests as possible of water flowing therein should be taken, giving the same results as obtain on smaller ships.

The Baule log directs the tracing of routes. To wire up the electro (13), Figure 1, in parallel for the maneuvering contact by the log, and to regulate the scale of the tracer's speed for the element of the course corresponding to the length of the tube.

The tracer of courses, we have said, is complete and certain. The log, on which the experiences are more recent, has not yet been tried on ships of all types, but the results obtained up to the present have certainly worked perfectly.

On board of the Cruiser 86, we have registered speeds between three and sixteen knots on various courses with a precision of 0.05 of a knot.

In salt water trials at Toulon, the error committed by the whole tracer of courses and the log bubble has not exceeded 100 meters.

This is a wonderful result, and the combining of these two apparatus suppresses completely errors of estimate and calculation and is about to render the most precious service to navigators.—"Le Yacht," translated by A. B. LeP. Mesny.

THE FLETTNER RUDDER ON MOTOR SHIPS.

A New Type to be Fitted on Three Motor Cargo Vessels.

The first real application of the Flettner rudder, the main object of which is to allow the rudder of a large ship to be operated manually without the necessity of employing a steering engine, was on the 8,000-ton motor cargo ship *Odenwald*, and particulars, both of the operation of the rudder and of the vessel itself, have already been published in this journal.

A further step is shortly to be made by fitting rudders of a modified design on three single-screw motor ships of 4,580 tons deadweight, which are being built by Krupp's, of Kiel, for Messrs. H. C. Horn, of Flensburg. These craft are 306 feet in length with a beam of 42 feet 6 inches and a depth of 25 feet 9 inches. A six-cylinder 1,400 B.H.P. Krupp four-cycle engine is to be installed, running at 125 R.P.M., and the speed is 10½ knots.

In the Flettner system there is a main and auxiliary rudder, the latter having a very small surface compared with the former and acting more or less on the main rudder in the same way as an ordinary rudder acts on a ship. In the latest form of the device to be fitted on the three vessels in question there are three rudder blades, as seen diagrammatically in Fig. 1.

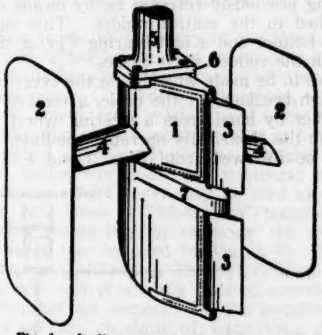


Fig. 1.—A diagrammatic illustration of the Flettner three-bladed rudder.

The main rudder (1) in the center is attached to the rudder shaft (5) by means of the flange (6), whilst two arms (4) serve to carry the side rudders (2). 3 is the auxiliary rudder, which is actuated from the spindle within the hollow rudder shaft (5) through horizontal levers enclosed in the casing (7).

It is claimed that, by the employment of a three-bladed rudder—if it may so be termed—increased steering capacity is obtained up to an angle of 45 degrees to 50 degrees, whereas with the single rudder the maximum is reached when the rudder is turned through an angle of 35 degrees. The height of each of the blades is less than with the single rudder to give the required blade surface. The result is that, even when the ship is light, the rudder is wholly under water, as indicated in Fig. 2, which gives an outline plan of one of the new ships and a sketch showing the after-end of a similar vessel fitted with a normal single rudder.

The arrangement of the installation of the rudder on the ship now being built is seen in Fig. 3. B is the middle rudder and C the two side rudders, D being the auxiliary rudder, which is actuated by the mechanism (O),

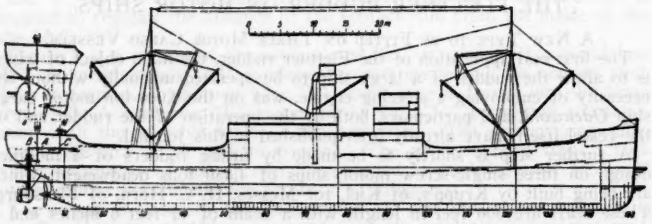
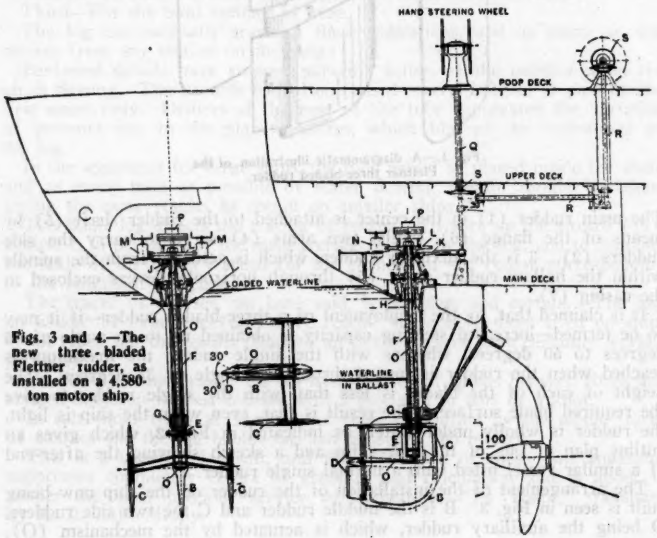


Fig. 2.—Profile of one of the motor ships to be fitted with the three-bladed Flettner rudder. Above is shown the stern of a ship with a normal rudder.

comprising horizontal links and a vertical spindle within the main rudder shaft. Steering is effected by means of a hand-operated wheel and a cable (not indicated on the drawing), which turns the spindle (O) through the intermediary of a pulley attached to it at the top in the casing P). At E is the flange coupling previously referred to, by means of which the rudder shaft (F) is attached to the central rudder. This shaft is carried in a bearing (G) at the bottom and a ball bearing (J) at the top, H being the gland through which the rudder shaft passes.

Arrangements have to be made whereby, in the event of the failure of the ordinary gear through breakage of the cable, a reserve means of operation can be provided, either by hand from a steering wheel at the after-end of the vessel or through the electrically operated windlass. The way in which this is effected can be followed from Figs. 3 and 4. The rudder spindle



Figs. 3 and 4.—The new three-bladed Flettner rudder, as installed on a 4,580-ton motor ship.

carries at its upper end a brake disc (L), which can be rigidly attached to a loose quadrant arm (K) by means of two brakes (M). These brakes are normally not in operation, so that the brake disc can move freely. If the auxiliary steering gear is required, the rudder brake (N) is first screwed down and then the brakes (M). The rudder can then be actuated as an ordinary fixed type, either by the steering wheel on deck or (through the shaft R) by the windlass.

The area of the central main rudder is about 39 square feet and the total area of the other two rudders is about 75 square feet, giving a combined area of 114 square feet, whilst the area of the auxiliary rudder is 6.6 square feet, the ratio being 1 to 17.

For the illustrations we are indebted to Werft Reederei, Hafen.—"The Motor Ship," Feb. 1924.

LIGNUM-VITAE OBSOLETE FOR STERN BEARINGS.

Introduction of New Rubber Bearing is Displacing Hard Wood and White Metal for Stern-Tubes on Small and Large Motor Craft.

Since Columbus first made his discovery of the Western Hemisphere, the product known as india-rubber has been of inestimable value to the progress of civilization, particularly since the advent of the internal-combustion engine. A Western inventor, C. Frederick Sherwood, a man whose personal experiences we have found most varied and interesting, recently came forward with an application of this substance that at first thought may seem impractical to the extent of being ridiculous, namely, bearings for stern-tubes of motorships, motor workboats and steamers; also for pumps, and for all bearing conditions where water may be used as a lubricant.

In the September 1922 issue of "Motorship" mention was made of the installation of a rubber stern-bearing replacing the usual babbitted bearing on a Diesel-powered tug designed by David W. Dickie for the State Harbor Commissioners of California. The installation of the rubber bearing at that time was a new and practically untried experiment in marine design. But it has given perfect success. On the installation being examined after ten months of service, no signs of wear were shown on either the bearing or the shaft. It has been in continuous use ever since, operating in the grit and sand of San Francisco Bay.

Since that time the exclusive manufacture and sale of these bearings has been taken over by the B. F. Goodrich Company, Akron, Ohio, who have been industriously exploiting their use in the marine field under the trade name Cutless. Numerous installations on the Atlantic seaboard on both pleasure and work craft have been made; one of the more recent ones being by the Standard Oil Company of N. J. in their new Diesel-electric driven barge No. 8, recently launched at Newport News, Va. The Standard Oil Company's engineers installed this rubber bearing after having operated a motor-barge in the Baltimore district under the most severe conditions, under which the bearings stood up perfectly.

While the actual use of rubber bearings in the marine field is only about two years old, these bearings have had four years of actual operation in the pump field. Such firms as the Worthington Pump & Machine Company, Chicago Pump Company and many others have adopted them as standard equipment on their pumps. It has been demonstrated time and time again that there is no substance that will stand the wear that rubber will. Rubber bearings have carried loads as high as 375 pounds per square inch projected bearing surface, and a wetted rubber surface has a lower co-efficient

of friction than an oiled babbit surface. Also, it has been absolutely demonstrated that no amount of grit or sand will either cut the bearing or score the shaft that operates in the bearing. This is due to the fact that it is impossible to imbed a piece of sand or grit in the soft rubber surface of the bearing, consequently, the sand or grit is rolled to a deep spiral groove in the rubber by the shaft, and is washed out by the water which also circulates through the grooves. Rubber bearings necessarily need water for their lubrication.

The spiral groove referred to runs the entire length of the bearing, and while providing a channel for the lubricant to circulate. also carries out any sand or grit which may come into the bearing. The surface of the rubber bearing being extremely resilient, the sand or grit cannot be imbedded in the actual bearing surface as just stated, but will roll to the groove, and be carried off. This action is unique only to a rubber bearing.

With a metal bearing the action is quite different. Upon the admission of sand to a metal bearing, in which the bearing surface is softer than the shaft which rotates in it, the sand becomes imbedded in the softer metal. The bearing then acts as a holding–tool for the sand or silicious matter, which then exerts a grinding or cutting action upon the rotating shaft.

It has been actually demonstrated in craft operation that the rubber bearing materially reduces screw vibration, cushioning any inequalities of pitch and not transmitting through the whole craft the entire vibration. This tends to longer life for machine operation and reduces the annoying screw vibration sometimes found in power craft. Rubber bearings in service are operating on speeds as high as 5,000 R.P.M. and as low as one revolution in three minutes.

It is a remarkable fact that misalignment of a shaft carried in a rubber bearing under high load is extremely small. Another feature of the rubber bearing permits a shaft that is misaligned to operate without pound or noise. A misaligned shaft operating in a metal bearing soon elongates the bearing. Permanent deformation of a rubber bearing is impossible, owing to the resiliency of the bearing surface. When operating on a misaligned shaft, the rubber bearing acts as a shock absorber, and prevents undue vibration being transmitted throughout the length of the shaft.

The Goodrich Cutless bearing as now on the market consists of a drawn–metal cylinder with longitudinal corrugations on the interior. Cement and a hard rubber coating are then placed next as a lining in the tube, followed with not less than $\frac{5}{16}$ of an inch of a specially compounded highly resilient Olivite rubber. The rubber is vulcanized to the shell under pressure. After vulcanization, the bearing is ground through with a high–speed grinding wheel. A speed of not less than 18,000 R.P.M., is found to be necessary in this connection. The spiral groove is also ground in the bearing during this operation. The finished bearing is then ready for installation.

The largest bearing operating to date, of this character, is three feet in diameter, by six feet length, and operates on a hydraulic turbine in the Mississippi River Valley. The introduction of these bearings to the marine field is rapidly progressing, and one of the foremost naval architects predicts that within five years the rubber bearing will replace the standard lignum–vitae bearing now in use on all craft.

Cutless rubber–bearings are made of particularly pure compounded. high–grade rubber stock, compounding of which must necessarily be extremely exact so as to permit of the longevity under all conditions. They are, we understand, actually guaranteed to give four times the life of any other bearing now being used in the marine field, including lignum vitae, which has a practically identical cost. The B. F. Goodrich Rubber Company

SHOWING THE RUBBER BEARING IN THE STERN TUBE OF THE NEW DIESEL-ELECTRIC TANKER BUILDING AT NEWPORT NEWS FOR STANDARD OIL CO. OF NEW JERSEY.

handles these bearings in over one hundred agencies in the United States. They are now manufacturing these bearings in France, and through their affiliations, in England and Germany.

In reference to the longevity of rubber under extreme wearing conditions, it might be mentioned that King George's court, London, is paved with rubber, and has stood thirty-eight years of wear. During the entire period, the maximum wear shown on this surface is only one-eighth inch.—"Motorship," Feb., 1924.

ECONOMY OF FUEL THROUGH INTELLIGENT OPERATION*.

By H. H. Norton, Lieut-Comdr. U. S. Navy.

Many times have I heard the question asked, "In what manner can the efficiency of a plant be most expeditiously increased?" Although there are numerous specific methods that might apply to a particular plant, there is generally one method that pertains to all. This is the saving that can be made as the result of intelligent operation in the generation of power necessary to operate the plant.

As the combustion of fuel applied to a boiler for the generation of steam is the most common method of supplying power for a plant, intelligent operation in firing in order to generate the steam is a subject of paramount import.

THE NAVY ESTABLISHES A FUEL OIL TESTING PLANT.

The Navy is a noteworthy example of an organization which has recognized the necessity of skillful operation in effecting economy. The advantages of fuel oil, particularly for marine purposes, could readily be seen, but the full benefits expected were not readily obtainable. To ascertain why they were not obtained, was the direct cause of the establishment by the Navy of a fuel-oil testing plant.

This plant was started about the time that the first oil-burning installations afloat were completed, but was not of great value until the year 1911, about a year after the operation afloat had shown its necessity. To remedy conditions afloat was its first mission, but later on it became an experimental plant for improvements as well as a testing plant of established methods.

It is interesting to note, among other things, that boiler efficiencies with oil-burning installations for the last ten years have, for the same rates of combustion, increased 10 per cent, and in some cases more than this. To date the highest boiler efficiency recorded at the fuel oil testing plant under test, *without* the use of preheated air, is very close to 84 per cent. Boiler tests in 1911 and 1912 show efficiencies ranging from 64 to 75 per cent with an average of about 70 per cent. While this increase in efficiency must be mainly accounted for by the improvements in apparatus, some of it should nevertheless be credited to more skillful operation.

EIGHTY PER CENT BOILER EFFICIENCY AT ONE HUNDRED PER CENT RATING.

As a result of experience with many types of boilers, I believe there is little excuse for operating in a manner that produces less than 80 per cent boiler efficiency at a 100 per cent boiler rating, nor should there be a drop of more than 1 or 2 per cent efficiency per 100 per cent increase in boiler rating. The operating engineer who is obtaining 80 per cent boiler efficiency is doing well and must observe his operation very closely for each

*Extract of paper presented before the Engineers' Club, Philadelphia, Pa., Jan. 15, 1924.

additional one per cent when striving for the maximum. But the one who is not obtaining 80 per cent could, with more intelligent operation, easily bring the efficiency to this figure.

It was the fact that results being achieved by the Navy under test conditions were not being duplicated in practice, that led the Navy to realize that the man who superintends or actually fires the boiler must be instructed as to the proper method. To accomplish this, competition and, later, education, were the means employed. Competition instilled a desire which at times proved very costly, until education was included.

The Shipping Board, following the steps of the Navy, is also steadily increasing efficiency in this line, and a larger percentage of this increase is attributed by them to the education of their operating engineers.

COST OF FUEL IS A MAJOR OPERATING EXPENSE.

Regardless of the type of plant maintained by any industrial concern, the cost of fuel is directly or indirectly a major operating expense. Fuel saved is money earned. But does the man who by his position is most concerned about the earnings of a company, know anything about the economical use of fuel? He is doubtless in a position to receive the ablest kind of engineering advice as to apparatus, but would he be inclined to spend money on the education of his operating force in such a manner as the Navy does? His adviser can probably state that he has seen certain results obtaind from an apparatus under test conditions, but when he tries to operate a duplicate installation he does not always reproduce the test results. When this occurs, is consideration given to the fact that the skilled personnel who conducted the original test were not in themselves duplicated?

What percentage of the cost of an apparatus is spent in educating its future operating force? A large percentage of your expenditures may go toward purchasing equipment and fuel, and if the opportunity arises to cut down the fuel bill a small percentage by purchase of additional equipment, it is generally done; but how much effort is spent with the means at hand through intelligent operation? On one hand, with skillful operation there is but a small percentage to work on to obtain the highest theoretical efficiency; whereas on the other, if operation is unskillful, there is a much larger percentage to work upon to achieve a saving at little expense.

By referring to the theory of combustion, it is not meant that it is necessary for every fireman to be educated to the point where he can figure out the boiler efficiency or strike a heat balance, but rather to where he knows the constituents of his fuel and the necessity for the combination of the combustible with just sufficient quantity of air to convert it completely to the gases that are the products of complete combustion. He should realize that the carbon, hydrogen and sulphur must receive sufficient air to be converted to carbon dioxide, water and sulphur dioxide, in order that the total number of potential thermal units in the fuel be abstracted; that insufficient air will cause the formation of carbon monoxide, and that when carbon monoxide is formed the amount of heat obtained from the carbon of the fuel is only about one-third of what it would be were the carbon converted to the dioxide.

He should know that smoke is unburned carbon, representing great loss of heat and consequently money—not only for the reason that the carbon leaving the furnace as smoke produces no heat and dirties the heating surfaces, but is also usually accompanied by carbon monoxide. He should also be cognizant to the fact that if more air is supplied than is absolutely necessary to convert the carbon to carbon dioxide, and the other constituent elements to their respective gases, the excess will dilute the hot furnace gases and it will be necessary to expend much heat to raise the

excess air to stack temperature; that the sooner a complete mixture of the fuel with air can be effected, the more chance the gas will have to give up its heat before escaping to the atmosphere; and that the lower the temperature of the gases escaping from the stack, the more heat has the boiler absorbed for the generation of steam.

These items together with knowledge of the instruments necessary to indicate the quality of combustion, should result in a majority of cases in a greater saving in the fuel bill than any other means.—"Power," Feb. 5, 1924.

THE GAS TURBINE.

In the author's opinion, while many attempts have been made to realize the ideal of an internal-combustion turbine, it may be briefly stated that nearly all of them have been failures or somewhat akin to failures. In fact, from his own experience with internal-combustion engines, the author comes to the conclusion that he is not convinced that even with the best results which have been obtained the gas turbine can be made a commercial success.

A few years ago the author was determined to set aside the sum of about £20,000 as a fund to be used for research and experimenting in gas turbines. This sum of money has not been used for this purpose, as he is convinced that at present there are almost insuperable difficulties in obtaining sufficiently satisfactory results to warrant such an expenditure.

The problem of the gas turbine is much more difficult than many people believe. In addition to the prime-mover element consisting of a rotating wheel, nozzles, combustion chambers, ignition, and valve gear, there must be an air blower or air compressor capable of feeding the combustion chambers of the turbine with sufficient air at suitable pressures for carrying on the combustion processes. There must also be a similar blower or compressor for delivering gas in a suitable volume and at suitable pressure for mixing with the air. There must also be suitable motors, whether of the internal-combustion or other type, for driving these blowers, and to obtain the maximum heat efficiency there must be some apparatus for utilizing the heat of the exhaust gases in some of the processes connected with the gas turbine. From this brief inventory it will be seen how many problems have to be solved before a successful gas turbine can be built.

The difficulties that have been experienced so far as are known have been mainly metallurgical, in that a metal has not yet been found which possesses high tensile strength at high temperatures. Herr Holzwarth has patented a pure mild steel, unalloyed as far as possible, with a carbon content preferably below but not exceeding 0.1 per cent for his turbine wheels, and although there are no practical data to confirm this statement, one infers that with material of this kind some success has been obtained. When one considers that the turbine disk, nozzles, and blades have to withstand a temperature of about 2,500 degrees F. and pressures reaching 300 pounds per square inch, it will be understood that practically a new material has to be found which will maintain not only its strength but also its form under the temperatures and pressures mentioned.

It is quite true that similar difficulties were experienced in steam-turbine blades and disks and that they have been largely overcome, but the temperatures even with highly superheated steam are only about one-fourth to one-fifth of that which is experienced in a gas turbine, so that the comparison is not quite a correct one.

The only gas turbine about which there has been any authentic information is that made by Hans Holzwarth. Sometime in 1923 the author

found references in the German press to a gas turbine which was invented by Leich of Hamburg. He put himself in communication with the inventor and was told that six turbines of this type were actually at work in Germany. He asked for further information and for permission to visit Germany and see the turbines at work but this was refused, even if a German engineer were employed by the author to investigate the motor. The inventor agreed, however, to come to England and show the drawings to the author. He informed him that he had a gas turbine of 12,500 horsepower working on blast-furnace gas in upper Silesia and that it had worked since July-August last, night and day, without a stop and that five other turbines had been built varying from 500 horsepower up The author expresses doubts as to the truth of these statements.

The Leich turbine, as shown in the drawings in the original paper, consists of a Laval rotor and nozzles combined with a labyrinth wheel and a great number of combustion chambers. The Holzwarth turbine has a maximum number of eight combustion chambers, whereas in the Leich turbine there may be as many as sixty combustion chambers round the circumference of the rotor according to the size of the machine.

The Leich turbine is constructed on the constant-pressure system, and in reply to queries as to whether there was any firing back when burning the explosive mixture, the author was informed that it had never occurred and that the labyrinth wheel prevented anything like that happening. The combustion chambers are extremely small in size, large ones not being necessary owing to the increased number of them, and on this form of construction Herr Leich places great importance.

Herr Leich informed the author that he guaranteed a thermal efficiency of not less than 36 per cent and that none of the turbines actually constructed gave less than 31 per cent. As the author could not be permitted either to enter into correspondence with the firms in Germany who were making the Leich turbine under license or to view any of the turbines said to have been constructed, he therefore declined all proposals made to him for manufacturing this machine, but as he has already indicated, there has been much newspaper publicity in Germany over this turbine, and as the way seems to be blocked for the ordinary engineer to make himself acquainted with what is going on, he repeats his suggestion that some of the enterprising engineering journals should find out the truth concerning this machine.

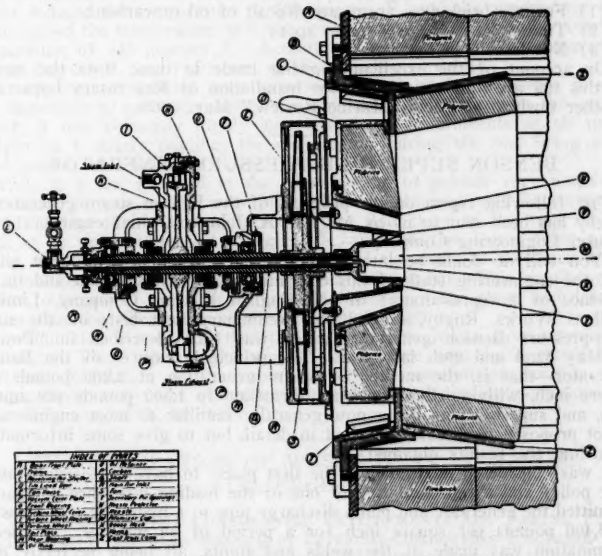
An illustration in the original articles shows a view of the Leich turbine. This is taken from a German magazine in which the turbine is described. A similar photograph was shown to the author by Herr Leich, and the former immediately said that the photograph was a faked one, which the inventor admitted was so, but he added the statement that it was purposely faked because he did not wish to show the whole of the construction of the turbine. (Hugh Campbell in a paper read Jan. 29, 1924, before the "Institute of Marine Engineers." Abstracted from advance publication, *d*)—"Mechanical Engineering," April, 1924.

THE RAY ROTARY FUEL OIL BURNER.

In 1915 William R. Ray of San Francisco was granted patents for an oil burner of the rotary type comprising many unique and valuable features.

The W. S. Ray Manufacturing Company was organized to produce and distribute this burner, and in the last eight years they have made many thousands of installations for heating buildings, for stationary power plants, and for industrial purposes. The engineers of the organization have accumulated a large volume of original data and information relative to the burning of fuel oil under boilers, and as a result have developed many exclusive refinements in design and application.

As will be seen from the sectional drawing reproduced herewith, the Ray rotary burner consists essentially of a hollow steel shaft with an air fan and an internal cone tip mounted at one end, and the rotor of an electric motor or steam turbine mounted at the other. The oil is introduced in a ⅜-inch pipe through the hollow shaft, this pipe being arranged to apply the oil on the surface of the internal cone at its smallest diameter.



SECTION THROUGH THE RAY ROTARY FUEL OIL BURNER OF THE STEAM TURBINE DRIVE TYPE.

In operation the hollow shaft with the fan and the burner tip are rotated clockwise at 3,000 to 4,000 revolutions a minute, so that the oil is flung from the outer edge of the surface of the internal cone in a whirling spray of finely atomized particles. Air is driven by the fan through the annual opening surrounding the atomizing tip, and in this opening spiral vanes are so disposed that this air is given a whirling motion counter clockwise. The air vaning is cut at different angles to meet different furnace conditions.

The effect of this combination is to throw a finely atomized spray of oil whirling at a peripheral velocity of about a mile a minute into fresh air whirling in the opposite direction at an equal or greater speed, with the result that practically complete combustion is obtained.

With the Ray system cold oil of 14 gravity can be burned—in fact the present heating now universally in use can be dispensed with entirely.

Oil pressure of 30 to 50 pounds is ample.

Cleaning or changing of tips is entirely dispensed with due to the large orifice in the Ray burner.

Forced draft is not necessary with Ray installation.

Quite recently an interesting test was made in a large tug on San Francisco Bay. This vessel was notorious as a smoke screen producer and for difficulty in maintaining steam pressure. With the Ray rotary burner installed no appreciable smoke could be observed and constant steam pressure was maintained with a marked improvement in economy of fuel. After 12 days' intermittent operation one of the boilers was opened for inspection and the following conditions observed:

(1) Furnace brick free from any deposit of oil or carbon;
(2) Tubes clean and free from soot;
(3) No defects in burners.

On account of the excellent showing made in these tests the owners of this tug are now considering the installation of Ray rotary burners on another towboat.—"Pacific Marine Review," Mar., 1924.

BENSON SUPER HIGH-PRESSURE GENERATOR.

THE following report on the progress of the Benson steam generator at Rugby has been sent to us by Mr. W. A. Johnston, chief engineer to the Benson Engineering Company:—

"You will no doubt be interested to learn that during the past week-end, the engineering staff of this company under my direction and in the presence of a representative of the English Electric Company, Limited, Willans Works, Rugby, carried out preliminary heat tests on the super high-pressure Benson generator, which was fully described in "Power" of May 22nd and 29th last year. The principal features of the Benson generator—that is, the method of steam generation at 3,200 pounds per square inch, without ebullition, then throttling to 1,500 pounds per square inch, and superheating—being now generally familiar to most engineers, it is not proposed to discuss the plant in detail, but to give some information regarding the results obtained.

It was considered advisable, in the first place, to have a covering insurance policy and a representative of one of the leading insurance companies submitted the generator and pump discharge pipe to a hydraulic test pressure of 6,400 pounds per square inch for a period of 20 minutes. A special examination was made of the welds and joints, all being perfectly dry, and with no signs of weakness anywhere. The hydraulic test therefore was very satisfactory, and the policy has been issued at a very low rate; in fact, at a rate certainly not more than that of an ordinary boiler rate.

The object of the present tests being only to prove the theory of generating steam without ebullition and to demonstrate the efficiency of the plant, and also to furnish data for the design of larger and more commercial units, the steam generator only was in operation and discharged directly to a surface condenser against a back pressure valve, which maintained a pressure in the superheater elements of about 1,500 pounds, special precautions being taken to exhaust to the condenser through an expanding nozzle and to cool partly the highly superheated vapor before it reached the condenser.

The initial pressure in the generator heating elements was also maintained constant by a back pressure valve between the generator and superheater, so that while the pressure in the generator was 3,200 pounds per square inch, the actual pressure in the superheater was 1,500 pounds per square inch.

The first heat test took place on Sunday, the 17th inst., when the plant actually operated under its own oil burners for about four hours, the object of that particular run being only to tune up, thoroughly dry out the brickwork and to make the necessary adjustments. The temperature of

the fluid was therefore intentionally limited to about 500 degrees F., the pressure of 3,200 pounds per square inch being maintained at all times.

A further heat test was carried out on Monday, the 18th inst., with the steam generator in commission about six hours, and the final results were that the pressure of 3,250 pounds per square inch was maintained in the generator heating elements at a temperature of 725 degrees F., the fluid on being throttled to 1,500 pounds per square inch in the superheater reduced the temperature to 620 degrees F., and the superheater elements raised the temperature at 1,500 pounds per square inch to an average temperature of 865 degrees F. Actually at certain periods a temperature of 910 degrees F. was registered at the superheater outlet.

No apparent difficulties were experienced, and while of course, it was not impossible to ascertain visibly the condition of the steam in the superheater, it was obviously highly superheated, the instruments at all times registering a steady reading, the evaporation during the test being over 8,000 pounds of water per hour.

While it is not intended at the present time to publish very much information, an idea of the very high efficiency obtained may be estimated from the fact that the temperature of the flue gases at the foot of the stack at no time exceeded 110 degrees F., the radiation losses being practically negligible, as the generator casing and the ducts to the stack were, by actual contact with the hand, only pleasantly warm. This result would appear to be one of the most remarkable features of the plant, and demonstrates that the efficiency must be exceedingly high.

I would also like to add that the high-pressure joints gave no trouble whatever. They have never been adjusted since first being fitted into position. Regulation of the superheat with a constant evaporation was comparatively easy, and altogether the test exceeded all expectations."—"The Engineer," Feb. 22, 1924.

THE UTILIZATION OF WASTE HEAT IN MARINE OIL ENGINES.

At the Present Time Considerable Developments are Taking Place in This Particular Branch of Internal-combustion Engineering. A Contributor Discusses Here the Theoretical Possibilities of the Idea.

Although the overall efficiency of oil engines is comparatively high, there is still a possibility of saving some of the heat exhausted by the gases to the atmosphere and utilizing it for various purposes. The amount of heat that can be saved is probably not big enough to justify elaborate and expensive arrangements. However, if it can be achieved by simple and inexpensive means, it is quite worth considering as will be seen from the calculations below. In general, if the thermal efficiency of an oil engine is high, less heat is wasted into exhaust, and thus a smaller percentage can be saved. Of course, it is more economical to utilize as much heat as possible in the engine by increasing its thermal efficiency, although this is not always possible. For oil engines working on a constant pressure cycle, thermal efficiency depends greatly on the adopted compression ratio, ratios of fuel cut-off, and exponents of the expansion and compression curves. The common formula for constant pressure cycle thermal efficiency is:—

$$(1) \qquad e_t = 1 - \frac{r^n - 1}{C^{n-1} n (r-1)}$$

where r is the ratio of cut-off, C compression ratio, n exponent of the polytropic curve for compression and expansion.

Cut-off is usually expressed in ratios of volume at the time the fuel valve closes to the clearance volume or, as may be seen on diagram 1,

$$r = \frac{V_r}{V_c} \tag{2}$$

The ratio of fuel cut-off depends chiefly on the quantity of fuel injected, provided the combustion is at constant pressure, and thus on indicated mean pressure. However, if the compression ratio were altered, the cut-off for the same indicated mean pressure would be different. This would be due to the difference in the thermal efficiency of the cycle with altered compression ratio, and a different quantity of fuel would have to be injected to obtain the same indicated mean pressure. Compression ratio is a very important factor in thermal efficiency, and its influence is illustrated in diagram 2, where a curve is given, showing the variation in theoretical thermal efficiency in a constant pressure cycle at a theoretical mean pressure of 115 pounds per square inch. The next important factor affecting thermal efficiency is the cut-off ratio. As has been stated, the cut-off ratio depends on the mean indicated pressure, provided the compression ratio is not altered. Thus, at different mean indicated pressures, the thermal efficiency is also different, *i.e.*, with the increase in mean indicated pressure thermal efficiency decreases. The influence of the cut-off ratio on thermal efficiency is illustrated in diagram 3, which is calculated for compression ratio, C=15 and exponent $n = 1 \cdot 3$. Curve 1 shows thermal efficiency for different mean pressures, curve 2 shows the ratio of cut-off, curve 3 shows the fuel consumption per horsepower-hour in pounds, curve 4 shows the temperature of the gas at the end of combustion, and curve 5 gives the percentage of air utilized for burning the fuel. It can be seen from this curve that at the above compression ratio all the air in the cylinder will be used at about 185 pounds per square inch theoretical mean pressure, which will correspond to about 170 pounds per square inch mean indicated pressure in a four-cycle engine, and about 157 pounds per square inch mean indicated pressure in a two-cycle engine.

HEAT WASTED IN EXHAUST GAS.

Speaking generally, the amount of heat wasted in the exhaust gas is usually inversely proportional to thermal efficiency. Therefore engines having lower compression ratios or running normally at a higher mean pressure are more suitable for waste heat saving installations. A very important factor for successful utilization of exhaust gas is the temperature of the gas in the exhaust pipe. As usually waste heat is utilized for raising steam in waste heat boilers, a greater percentage of heat can be utilized in a boiler if the initial temperature of gas is higher. This is evident from the calculation given below. The percentage of heat utilized in the boiler of the total carried by exhaust gas equals:—

$$H_1 = \frac{t_1 - (t_2 - t_o)}{t_1} e \tag{3}$$

where t_1 is the temperature of exhaust gas, t_2 the temperature of water or steam in the boiler, t_o the temperature difference between exhaust gas leaving the boiler and the temperature of steam, and e boiler efficiency, which includes radiation only, as the other factors are already taken into account in equation (3).

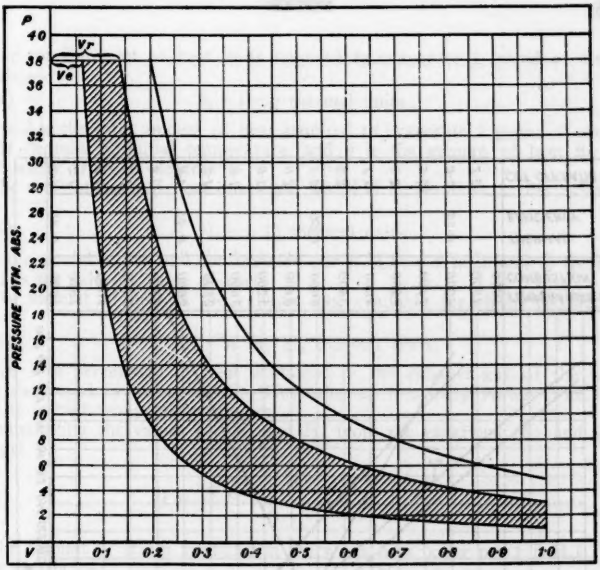


Diagram 1.—Showing the way in which Cut-off may be expressed in Ratio of Volume at the time the fuel valve closes, to Clearance Volume.

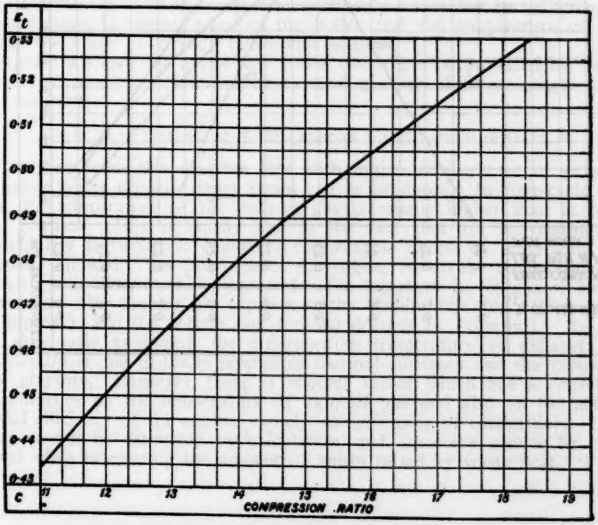


Diagram 2.—Illustrating the effect of Compression Ratio upon Thermal Efficiency.

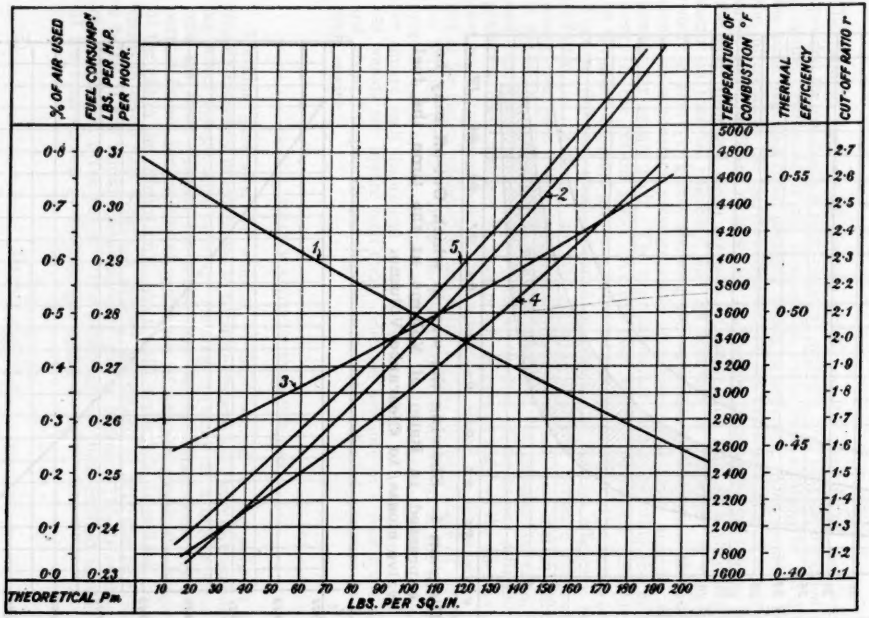


Diagram 3.—Illustrating the influence of Cut-off Ratio upon Thermal Efficiency.

The total amount of heat units required to evaporate 1 pound of steam equals:—

(4) $$h_1 = l - g \text{ thermal units,}$$

where l is the total amount of heat required to evaporate 1 pound of water of 32 degrees F. initial temperature, and g is the amount of heat in the feed water from 32 degrees F.

The total amount of heat developed per B.H.P. per hour in the cylinder is:—

(5) $$H_2 = q\ Q \text{ thermal units,}$$

where q pound is the fuel consumption per B.H.P. per hour and Q thermal units is the calorific value of fuel.

The amount of heat in the exhaust gas, available for steam production, is:—

(6) $$H = h\ H_1\ H_2 \text{ thermal units,}$$

where h is percentage of heat contained in the exhaust gas of the total heat developed in the cylinder. The value of h usually varies from 0.26 to 0.32 in high compression engines.

Substituting the values for H_1 and H_2 from the equations (3) and (5), we get:—

(7) $$H = \frac{h\,q\,Q\,[t_1 - (t_2 - t_0)]}{t_1}\,e.$$

The amount of steam produced in the waste heat boiler per 1 B.H.P. per hour equals:—

(8) $$W = \frac{H}{h_1} = \frac{h\,q\,Q\,[t_1 - (t_2 - t_0)]}{(l - q)\,t_1}\,e.$$

Boiler efficiency e in this case represents only losses due to radiation, as no combustion is taking place in the boiler and the temperature of gas leaving the boiler is already taken into account.

Diagram 4 shows the amount of steam which can be produced by waste heat for different temperatures of exhaust gas and different steam pressure, assuming:—

$$h = 0.30,\ q = 0.43,\ q = 30 \text{ (approx.) and } t_0 = 90 \text{ degrees F.}$$

We can see from this diagram that with higher temperature of exhaust and lower boiler pressure more steam can be produced. In four-cycle engines, the temperature of the exhaust gas is usually higher than in two-cycle engines. This is chiefly due to the fact that for scavenging of the cylinders in two-cycle engines a considerable excess of scavenging air is usually allowed. The influence of scavenging air on the temperature of exhaust gas is illustrated in diagram 5. The top curve is for gas of initial temperature 950 degrees F. (approx.), the middle one for 750 degrees F. (approx.), and the bottom one for 550 degrees F. (approx.). As has been previously explained, the difference in temperature of exhaust gas theoretically is due to the difference in thermal efficiency and the indicated mean pressure. However, there is another factor which has a considerable influence on the temperature of exhaust gas and also on the actual thermal efficiency of the engine, namely, the efficiency of combustion. This factor cannot be estimated by calculation, and therefore cannot be considered when calculating the amount of steam raised by waste heat.

upon Thermal Efficiency.

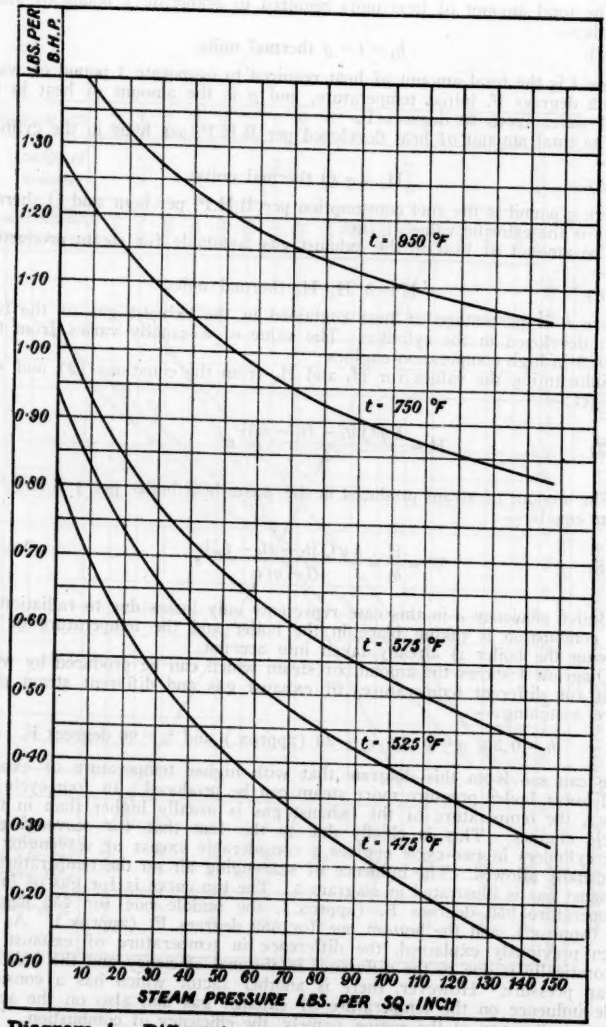


Diagram 4.—Different amounts of Steam produced for varying Exhaust Gas Temperatures and Steam Pressures.

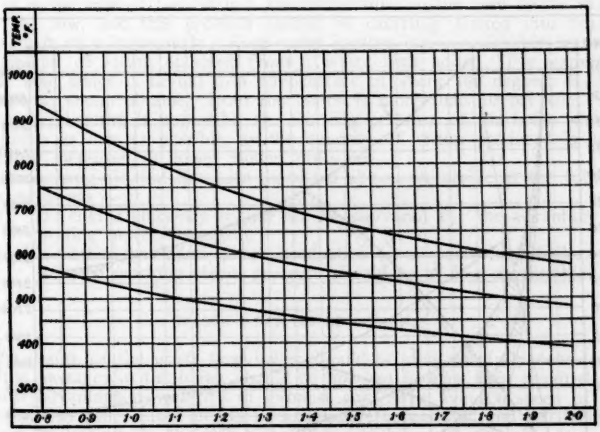


Diagram 5.—Showing the Influence of Scavenging Air on the Temperature of Exhaust Gas.

COMBUSTION EFFICIENCY.

When the engine is well tuned and the fuel valves are clean, combustion is usually more efficient; but as soon as the sprayers wear, the injection pressure drops, or the fuel valve lift is altered, the efficiency of the combustion becomes less. Inefficient combustion can usually be noticed on the indicator diagrams. A low value of the exponent for the expansion curve in an indicator diagram is a sure sign of after-burning, which is the result of inefficient combustion. In every indicator diagram we can easily find traces of after-burning, but at high mean pressures it becomes more pronounced. In extreme cases of after-burning, when the fuel is burning during a considerable part of the expansion stroke, the efficiency may drop and the temperature of exhaust gas increase considerably. The effect of after-burning is illustrated on diagram 6. Curve 1 is the compression line. curve 2 is the theoretical expansion line, corresponding to a theoretical mean pressure of about 115 pounds per square inch, as shown on diagram 1. If the same quantity of fuel were burning slowly during the first half of the expansion stroke, the combustion curve would take the shape of curve 4. The theoretical mean pressure in the latter case would drop to 69 pounds per square inch, and thermal efficiency would come down to 0.285, instead of 0.48 with normal combustion. To obtain the required theoretical mean pressure of 115 pounds per square inch we shall have to burn more fuel in the cylinder.

Calculation shows us that the amount of fuel required to produce 115 pounds per square inch theoretical mean pressure, with the slow combustion taking place during the first half of the expansion stroke, would be about the same as is required with efficient combustion to produce theoretical mean pressure of about 185 pounds per square inch. Curve 5 for slow combustion corresponds to about 115 pounds mean pressure; of course, the temperature of exhaust in the latter case will also correspond to the

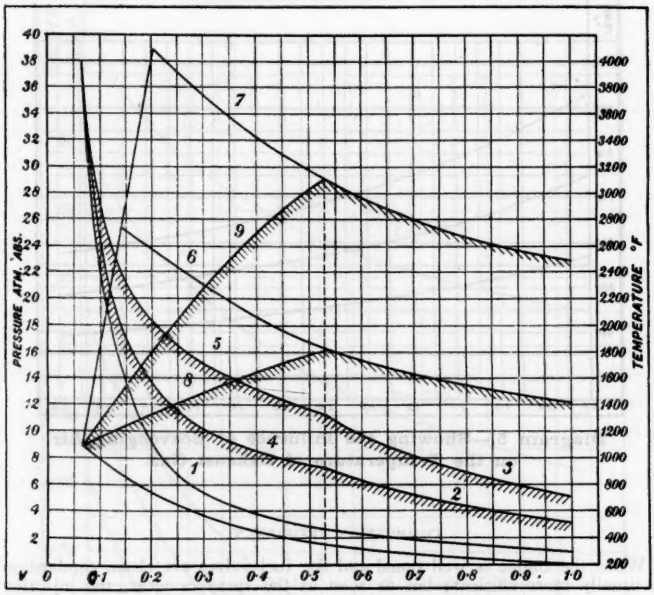


Diagram 6.—Showing the Effects of After-burning.

amount of fuel injected, and will be considerably higher. Curves 6 and 7 give the temperatures for normal combustion and expansion; curves 8 and 9 for slow combustion, corresponding to pressure curves 4 and 5. The calculation for the above curves is very complicated, and is not given here, as it has no direct connection with the subject of this article. However, the above example shows the importance of good combustion and also the influence of slow combustion on the amount of waste heat in the exhaust. At the end of a long voyage the quantity of heat carried away in the exhaust usually increases slightly for the reasons explained above, and this leaves a certain margin for the benefit of waste heat boiler.

AFTER-BURNING.

For the safety of the engine, bad combustion or after-burning is very detrimental, as it increases the fuel consumption and the quantity of heat developed during combustion period, and also the heat absorbed by the cylinder and cylinder-cover walls, thus increasing the heat stresses. From the equation (8) we can see that heating of feed water in a waste heat boiler will be an advantage. As the exhaust gas leaving the boiler still has a fairly high temperature, it can be further utilized for heating the feed water. However, there may be a danger of condensation of sulphuric

vapors in the case of the exhaust gas temperature in the feed heater dropping too low, and this problem should be carefully looked into before applying it on a large scale. Feed-water heating can considerably increase the amount of steam obtained from a waste heat boiler. For example, if the feed water is heated to a temperature of about 160 degrees F., the increase in steam obtained from the boiler is more than 10 per cent. If feed water heating is impossible the working pressure of the boiler should be chosen as low as possible, as the quantity of steam produced is considerably increased at lower boiler pressures.

Summarizing all that has been discussed above, we arrive at the following conclusions:—

(1) The engine efficiency should not be sacrificed for the advantage of getting more waste heat.

(2) To obtain maximum steam production in a waste heat boiler, the adopted working steam pressure should be as low as possible.

WASTE HEAT BOILER DESIGN.

The design of the waste heat boiler should be very carefully considered. All the passages in the boiler should be not less in area than required for a most advantageous velocity of exhaust gas. The average speed of exhaust gas in boiler tubes should be somewhere between 80 and 100 feet per second, reckoning on the volume of gas at the average temperature of exhaust in the boiler. The temperature of exhaust gas usually varies from 950 degrees F. to 750 degrees F. in four-cycle engines, and from 600 degrees F. to 500 degrees F. in two-cycle engines. Therefore a larger quantity of steam can be produced in a waste heat boiler with a four-cycle engine than with a two-cycle one. The passages in the boiler and the exhaust pipe should be calculated for the average velocity of gas in the passage, considering the flow of gas to be continuous.

On the above basis, the area of the passages in the boiler, in square feet will be:—

$$A = \frac{W}{V} \text{ square feet} \tag{9}$$

where W is the volume of gas exhausted per second. V the average velocity in feet per second.

Volume of gas in a two-cycle engine is:—

$$W = \frac{D^2 \times S \times Z \times R \times m \times T_e}{71{,}000{,}000} \text{ cubic feet} \tag{10}$$

and in a four-cycle engine:—

$$W = \frac{D^2 \times S \times Z \times R \times T_e}{167{,}000{,}000} \text{ cubic feet} \tag{11}$$

where D and S are working cylinder bore and stroke in inches, Z number of cylinders, R revolutions per minute, T_e degrees F. the absolute temperature of exhaust gas in the pipe, and m the ratio of volumes of scavenging cylinder to the working cylinder. The suitable gas velocity can be taken from 80 feet per second to 150 feet per second, the smaller velocity for two-cycle engines, and the higher for four-cycle engines.—"The Marine Engineer and Naval Architect," March, 1924.

NOVEL FEATURES OF A LARGE TURBINE.

A 30,000–KW. turbo-generator, the largest machine so far manufactured in Great Britain, is described in "Engineering" of Jan. 18 and Feb. 1, 1924. This is designed for 200 pounds gauge, total temperature 588 degrees F. and vacuum 28½ inches, giving a water rate of 11.1 pounds at 25,000 KW., built by the British–Thomson–Houston Co., of Rugby, England. This contains one velocity stage two-row wheel and 13 Rateau stages.

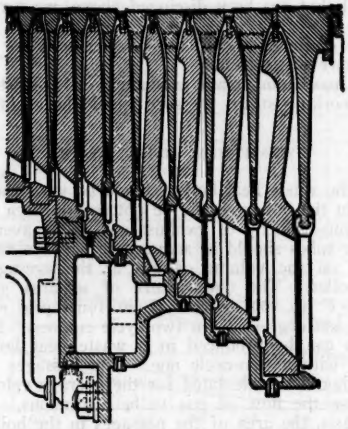


Fig. 1—Water extracted in last six stages

An interesting means of draining water from the lower stages of the cylinder is shown in Fig. 1. This represents a section of the lower wheel casing. It will be seen that each diaphragm contains a groove to catch water that is thrown off the blade by centrifugal force. Each of these grooves is drained at the bottom of the turbine casing.—"Power," April 8, 1924.

OIL FROM COAL.

Little has been heard lately of the possibility of producing oil in bulk by the distillation of coal, although it is generally known that both in private firms and at the laboratory of the National Fuel Research Board experiments are being carried out with a view to developing a process which would be successful from the commercial point of view. Particulars have recently appeared in a contemporary giving the average results of the operation of a low-temperature coal-carbonization plant for a continuous run of 128 days, in which it is stated that the average product from each ton of coal carbonized was: 15 hundredweight of smokeless fuel, 20.9 gallons of oil, 40 therms of gas, and 20 pounds of ammonium sulphate. An interesting feature of this report lies in the statement that the oil obtained

is "better than Diesel oil." It has been stated in the past that the oil obtained from coal, while it may be suitable for burning in the furnaces of a steam boiler, is hardly suitable as a fuel for Diesel engines, but according to the report under notice this difficulty has now been overcome. Another feature of interest is that the smokeless fuel has been sold at prices equal to that of the best domestic coal. There is, of course, still the difficulty of creating a large market for this fuel, but, provided this can be done and the price obtained for the fuel is high, it should be possible to sell the oil at a figure which should encourage its use for internal-combustion engines.—"Shipbuilding and Shipping Record," Jan. 1924.

LUBRICATION OF REFRIGERATION COMPRESSORS.

By Allen F. Brewer.

While anhydrous ammonia is perhaps the most generally used refrigerant today, there are a number of other chemicals, such as carbon dioxide, sulphur dioxide, propane, ethyl chloride and methyl chloride, which can be more or less readily employed for the purpose of refrigeration, with certain decided advantages. In fact, from the viewpoint of safety in handling many of these are often regarded as preferable to ammonia. Therefore, the lubrication requirements of their respective machinery will be of interest, since these involve certain marked differences from the lubrication of the ammonia compressor.

CARBON DIOXIDE AS A REFRIGERANT.

Carbon dioxide (CO_2) or, as it is often termed, carbonic anhydride, is perhaps more like ammonia in its application to refrigeration purposes than any of the other lesser refrigerants. In fact, the only difference in machine requirements is that more rigid construction of the compressor is necessary, for the pressures (250 to 1,000 pounds per square inch) are considerably greater. The natural advantages of carbon dioxide over ammonia as a refrigerant, however, outweigh this possible disadvantage, in the opinion of many authorities, for it is non-explosive, non-poisonous, non-combustible, odorless, cheaper, neutral in chemical reaction and an effective fire extinguisher. As a result leaks in a carbon-dioxide system may be often disregarded and never feared, even though they may possibly occur more frequently on account of the higher operating pressures involved.

An added advantage exists by virtue of the fact that straight mineral oils have practically no affinity for carbon dioxide. As a consequence the possibility of the compressor oil being carried over into the refrigerating system by the compressed gas is negligible. In fact, this could occur only in event of the oils becoming vaporized, and the compressor discharge temperatures will never be high enough to promote any appreciable vaporization of the lubricating oil. Suitable oil traps should be installed in the compressor discharge line to catch any oil that might pass over with the refrigerant in the form of spray.

The most suitable lubricant for a CO_2 compressor is a straight mineral product of fairly low viscosity. It should possess two essential characteristics, a low pour test and a flash point sufficiently higher than the customary discharge temperature to insure that it will not be subjected to vaporization or distillation.

Adequate sealing of the stuffing boxes in a CO_2 compressor, whether vertical or horizontal, is a decidedly important matter. In general stuffing

boxes are sealed, piston rods lubricated, and leakage of gas prevented by force-feed lubrication, the lubricant being fed through an oil lantern or hollow space located between two separate sets of rod packing.

SULPHUR DIOXIDE USED IN SMALL SYSTEMS.

Sulphur dioxide is another chemical that is somewhat used in small refrigerating systems. It is not as satisfactory as carbon dioxide owing to its tending to show an acid reaction when in solution with water and the fact that it is a fairly active poison in gaseous form. It is not, however, regarded as dangerous to use on this account, any more than is ammonia. The outstanding feature in the application of sulphur dioxide to refrigeration purposes is the low pressures in the system, but the fact that it requires a relatively high compressor volume has largely interfered with its extended use.

Lubrication of sulphur-dioxide machinery involves certain difficulties due to the inherent nature of the refrigerant itself. In the first place the lubricant must be absolutely water-free, otherwise the gas will tend to react with this water to form sulphuric acid, which would prove very destructive to the metals from which the compressor is built. The effect that sulphur dioxide may have upon the lubricant must also be considered. Mineral oils as usually obtained have not proved to be absolutely satisfactory lubricants owing to the fact that sulphur dioxide tends to manifest an affinity for certain of the hydrocarbon constituents. As a result many mineral oils will be altered both chemically and from a lubricating point of view, to a certain extent. Some authorities consider it quite possible, however, to take mineral oils of somewhat higher viscosity and so refine them by either filtration or suitable chemical treatment, that their hydrocarbon components which may be affected by the sulphur dioxide, will be removed. The resultant product is then claimed to be an efficient and dependable lubricant.

In ordinary operation, however, sulphur-dioxide refrigerating compressors are usually lubricated with chemically pure glycerin or glycol. The latter is frequently spoken of as being perhaps the most effective sulphur dioxide compressor lubricant on account of its relatively low pour test, especially when it has been compounded with deflocculated graphite.

ETHYL AND METHYL CHLORIDES.

Ethyl and methyl chlorides can be discussed together from the viewpoint of the lubrication requirements of their systems of refrigeration, because of their similarity. In general they will be found to show no reaction with air or water, nor are they poisonous or objectionable as to odor. In addition, they are chemically neutral and only slightly inflammable. They have, however, relatively low specific heats, and larger volumes of each are required for the development of a given amount of refrigeration as compared to the volume of ammonia. So the rotary type of compressor is preferred by many owing to its ability to handle effectively large volumes at low pressures. On account of the low pressures involved, the loss of lubricant *via* stuffing boxes, etc., is generally smaller than where high-pressure refrigerants are used.

Lubrication of compressors wherein these refrigerants are to be used must be considered from the viewpoint of the effect the refrigerant will have upon the lubricant. Mineral oils are generally regarded as being unsuited, owing to the fact that certain hydrocarbons are more or less soluble in these chlorides. Chemically pure glycerin is, therefore, commonly used as the lubricant. The glycerin must be free from water or other foreign matter, otherwise poor lubrication and a certain amount of gumming will take place. In fact, the builders of compressors for such service

recommend only the use of chemically pure glycerin at all times. In using or storing glycerin it must be remembered that it has an affinity for water and tends to absorb moisture from the air when exposed. It should, therefore, be kept as nearly air-tight as possible at all times.

The cycle that glycerin follows in performing its lubricating functions in a rotary compressor is of interest. In general, the start of the cycle is its delivery to each end of the compressor *via* suitable lubricant connections and through ducts or channels to the bearings, passing along these to the interior of the compressor. It is then discharged to the condenser and separator along with the compressed refrigerant. Here separation of the two is effected by virtue of the lubricant being the heavier, the refrigerant passing on to serve its intended purpose, the lubricant being forced back to the compressor for re-usage. This is a continuous cycle during operation. Generally, a suitable strainer and sight-feed device is installed to insure the maintenance of effective and sufficient lubrication at all times.

PROPANE AND OTHER HYDROCARBONS.

While the idea of using certain hydrocarbon gases such as propane, ethane and butane for the purpose of refrigeration is not new; the actual application on a commercial scale is relatively recent. Propane has taken precedence over the others and is today a feature in certain household and commercial refrigerating systems. Propane is of advantage in that comparatively low temperatures are attained by its usage; it also requires lower pressures for compression. Furthermore, it is non-corrosive, involves no acidic reaction with water, is relatively stable under usual refrigerating conditions, is not dangerous except in event of sufficient concentration to exclude oxygen and is adapted to usage in a standard type of ammonia compressor if desired. On the other hand, as a hydrocarbon it is inflammable; therefore, open flames should never be allowed in its vicinity.

Satisfactory lubrication of refrigeration machinery wherein propane is employed is as yet in more or less of a development stage. As in the case of ethyl chloride a lubricant must be selected no part of which will be soluble to any extent in the refrigerant, otherwise its lubricating ability will be impaired more or less seriously. Mineral lubricating oils are, therefore, not regarded as suitable by many authorities. Experiments have been carried out using castor oil, lard oil and certain glycerides. As far as is known, these products or certain of their compounds have indicated their suitability where normal operating conditions exist.—"Power," April 1, 1924.

REINFORCED GRAY-IRON CASTINGS.

Description of a process that has been successfully employed for the manufacture of automobile-engine cylinders by a Belgian foundry.

Every foundryman who has had experience with casting automobile-engine cylinders knows how difficult it is to satisfy modern specifications which demand at the same time thin walls, intricate castings, and a leak-proof metal. In this case all the parts of the cylinder where porosity is frequent (valve seats, spark-plug ports, etc.) are provided with a thin steel wall which welds closely to the iron and insures perfect tightness. These walls are made of extra mild sheet steel which is given the proper shape before being introduced into the mold. The steel reinforcement (Fig. 1) is placed into the mold in such a manner that it welds to the exterior surface of the cylinder tube. The thickness of the reinforcing sheets has to be properly computed with due regard to the thickness of the cylinder walls in

such a manner that the cast-iron mass will weld to the steel without absorbing it completely. In the last eighteen months a number of cylinders so cast have proved that by this process porosity may be completely eliminated.

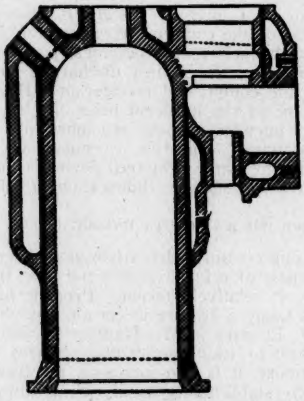


FIG. 1 MEYER METHOD OF CASTING REINFORCED CAST-IRON CYLINDERS

Because of the use of this method of preventing porosity in castings, it becomes possible to cast the cylinders in soft graphitic irons. The same process has been employed for making high-pressure gas and liquid containers. Abstracted from an article entitled La Fonte Armée, in "La Fonderie Moderne," vol. 18, Jan., 1924, p. 16, 1 fig., *d*).—"Mechanical Engineering," April, 1924.

THE LARGEST STEAMER CONVERSION.

GEARED TURBINES REPLACED BY A 3,000 I.H.P. SUN-DOXFORD ENGINE IN A VESSEL OF 11,620 TONS D.W.

The motor ship *Challenger*, which recently started on her maiden voyage around the world under charter to Norton, Lilly and Co., New York, is the largest steamer to be converted to a motor ship. She carries 11,620 tons deadweight, and is 410 feet in length with a beam of 56 feet, a depth of 41 feet, and a loaded draught of 30 feet 6 inches. She was built in 1918 by the Bethlehem Shipbuilding Corporation for the U. S. Shipping Board, and was purchased for conversion to oil-engine drive by the Sun Shipbuilding and Dry Dock Co. A 3,000 I.H.P. Doxford engine built by the Sun Co. is installed, running at 77 R.P.M. The propelling engine and auxiliaries were fitted in the space formerly occupied by the steam boilers and turbine, which was a 2,400 S.H.P. Curtis type running at 3,500 R.P.M., with gearing reducing the speed to 90 R.P.M.

As electrical auxiliaries have been installed on the *Challenger* instead of the original steam plant, the following details of this machinery will be of interest, since in many conversions the steam auxiliaries have been retained. The generating plant comprises three 65-K.W. dynamos driven by a new type of Worthington two-cycle two-cylinder engine with cylinders 12½ inches by 13¼ inches. There are two auxiliary air compressors direct coupled to two of the generating engines, each with an output of 150 cubic feet of air per minute at 700 pounds per square inch. They are of the two-stage type.

The remaining auxiliaries are as follows:

One Mianus oil-engine-driven emergency lighting dynamo set. One electrically driven two-stage Worthington emergency air compressor. One electrically driven horizontal duplex Worthington fresh-water pump, 3¾ inches by 4 inches. One electrically driven horizontal duplex Worthington bilge pump, 6 inches by 6 inches. Two electrically driven Worthington rotary lubricating oil pumps, 150 G.P.M. One 3-inch electrically driven centrifugal Worthington oil-cooler salt-water circulating pump. One electrically driven two-stage centrifugal Worthington 3-inch general-service and sanitary pump. Two electrically driven horizontal duplex Worthington distilled-water-circulating pumps, 8½ inches by 12 inches. One electrically driven Sun-Doxford auxiliary fuel pump. One electrically driven centrifugal Worthington 5-inch salt-water circulating pump. One electrically driven horizontal duplex Worthington bilge, ballast and fire pump, 8½ inches by 12 inches. One 150 G.P.M. electrically driven rotary Worthington fuel-oil transfer pump. One 15-K.W. motor generating set. One donkey-boiler, 150 square feet heating surface. One donkey-boiler feed pump (steam), and one donkey-boiler fuel-oil service pump.—"The Motor Ship." April, 1924.

THE TENDENCY OF MARINE OIL ENGINE DESIGN.

In the early days of the marine oil engine the would-be-manufacturer was faced with the problem of whether to adopt the two or the four-stroke cycle in his design. Of the two types the four-stroke presented much the easier problem to the designer, and opinion leaned towards this type. This was largely because many valve scavenging two-strokes were built, whose complicated cylinder head, subject to much greater heat stresses than that of the four-stroke engine, was a constant source of trouble. With the introduction of satisfactory port scavenging systems, the problems of design of the two-stroke engine became much simpler, and it is significant to note that at the present day most designs of marine engines manufactured in this country operate on the two-stroke cycle. Only those firms who gained a reputation in the early days of the four-stroke engine are still able to obtain orders for that design. It is the shipowner who ultimately settles the type of engine that is to be adopted, and there can be little doubt that mechanical simplicity and low first cost are the main points that he looks for when he is considering the possibility of adding oil engine propelled vessels to his fleet. To reduce further first cost and to increase the maximum power per cylinder there has been a tendency during the last few months towards the design of double-acting engines; many of these have, however, been mechanically too complicated to appeal to the shipowner. One of the simplest double-acting engines is that of the M.A.N. Company which was described in the German technical press last year. This engine has port scavenging and only fuel and starting valves in the cylinder cover. There are those who believe that the future oil engine will be a double-acting two-stroke with port scavenging, as this will give an engine of low

first cost and great mechanical simplicity. Another direction in which design is tending towards greater simplicity is in the method of fuel injection. Here we find that the air injection system gained almost universal support in the early days of the Diesel engine because, with it, good combustion could easily be obtained. Now, however, it has been proved by experiment that equally good results can be obtained with solid (or as it is sometimes called mechanical) injection of the fuel. There is in fact a movement towards the general adoption of this system because of the great reduction in first cost that is obtained by its use. Solid injection systems are already in regular use on the Vickers, Doxford and Still engines.—"The Marine Engineer and Naval Architect," Feb., 1924.

SEMI-STEEL CHARACTERISTICS.

There are many requirements in the several engineering industries which, while not demanding a metal possessing the high tensile properties of steel, nevertheless call for a material representing greater properties of ductility and homogeneity than cast iron, and which, in addition, may be capable of being hardened and highly polished. Writing recently in the "Foundry Trade Journal," Mr. David McLain described some interesting properties of a metal which would appear to meet the requirements already outlined, and which he claims to have manufactured to his own formula as far back as 1903. This metal he called "semi-steel," and it is stated that semi-steel exceeds, both in temperature and fluidity, any other mixture melted in the cupola. For its manufacture no extra-coke fluxes or new equipment are necessary. The secret of its successful manufacture is the casting temperature, and unless the melting and pouring temperatures are properly attained. good semi-steel, in the author's opinion, will never be produced, no matter how much attention may be paid to the question of mixture. Semi-steel is very homogeneous; it will twist or bend, and may, in addition, be hardened and highly polished. Its tensile strength is of the order of 25 tons per square inch, but, unlike alloy steels, it is not capable of withstanding shocks. During the late war many United States Shipping Board vessels were fitted with semi-steel propeller blades, mounted on a cast-iron boss. The inability of this material to withstand the heavy shocks associated with a "racing" propeller under light ship conditions resulted in many broken blades, and latterly the practice of fitting semi-steel blades was officially discouraged.—"Shipbuilding and Shipping Record," Jan. 2, 1924.

FILTERING DIESEL OIL.

In the lubricating oil system of any type of marine engine, particularly where forced lubrication is employed, special precautions are taken to ensure that the oil entering the bearings shall be entirely free from any foreign matter, and to this end the oil is carefully strained and filtered. The reasons for this are obvious, although it is not usually recognized that they apply almost as powerfully to the fuel oil in the injection system of a Diesel engine. Wherever oil—or any other fluid—passes at high velocity over a metal surface, the presence of the smallest particles of solid matter will lead to scoring of that surface, and thus at the fuel valves, if oil is used which has not been thoroughly filtered, the result will be a rapid scoring of the surface of the valve, or of the surface of the seating, which will lead to ineffectual closing with its consequent evils. In Diesel engines working on

the solid or airless system of fuel injection, this effect will also be noticed at the fuel oil pumps, and it will, of course, be accentuated throughout the entire system by the fact that such high working pressures are employed. We heard recently of a vessel on which the only trouble experienced in the running of the Diesel engines was due to leaky fuel-pump valves, the seats of which were so rapidly scored that frequent grinding in was necessary, and it was recognized that the trouble was due to the traces of solid matter in the Diesel oil. It is apparent, therefore, that careful filtering is essential in the fuel oil system.—"Shipbuilding and Shipping Record," Jan. 2, 1924.

A DIRECTIVE TYPE OF RADIO BEACON AND ITS APPLICATION TO NAVIGATION.

By F. H. Engel and F. W. Dunmore.

[ABSTRACT.]

The problem of improving the safety of marine and aerial navigation in time of fog and poor visibility has always been an important one. Many aids to such navigation have been established, one of the latest and most effective of which has been the radio direction-finder installed on shipboard and used in conjunction with radio beacons installed on light vessels.

This paper describes another method of safeguarding marine and aerial navigation, and, although somewhat restricted in its applications, the simplicity of the method makes it of special value in many instances. It makes navigation between fixed points possible regardless of visibility conditions and without dependence on landmarks or the magnetic compass.

The method requires no apparatus other than an ordinary receiving set on the ship or airplane. The directive transmitting properties of coil antennas are used which serve as a particular type of radio beacon. A coil antenna is directional when used as a transmitting antenna, much as it is when used as a receiving antenna; that is, the familiar figure-of-eight characteristic is obtained. This property is made use of in the directive type of double-coil antenna beacon described.

Two transmitting coil antennas are arranged at an agle of 135 degrees with respect to each other. These coil antennas are arranged to be connected alternately to a transmitting set by means of a special switch, which is thrown rapidly from one closed position to the other. Waves are thus intermittently propagated directively from each coil, the intensity of the emitted wave with respect to the plane of the coils varying in accordance with a figure-of-eight characteristic. Anywhere along the bisectors of the angles formed by the two coils a receiving set will receive signals of equal intensity from the two coils. At any point not on one of these bisectors, signals from one coil will be strong and those from the other coil weak, giving an inequality in the two signals which may be readily noticed, since one signal occurs directly after the other.

Experiments conducted at the Bureau of Standards are described in which two single-turn coil antennas are used crossed at an angle of 135 degrees. These coils were 45.75 by 15.25 meters (150 by 50 feet) in size and were connected alternately to a 5-K.W. quenched-spark transmitting set. A frequency of 300 K.C. (1,000 meters) was used. Receiving observations were made on the U. S. Lighthouse Tender *Maple* along a course which extended east and west at a point on the Potomac River about 56.4 K.M. (35 miles) south of Washington. At this point there was found to be a

zone of equal signals about a mile wide, and in which no noticeable difference in intensity of the signals received from the two coils could be noticed. Outside this sector, however, the two signals were noticeably unequal in intensity.

Interesting results were obtained as a result of tests conducted on an airplane. Recent tests with a short trailing wire and heavy weight have shown this method to be effective in enabling a pilot to guide his airplane into Dayton, Ohio, from a point 161 K.M. (100 miles) distant.

One of the great advantages of this system over methods employing a radio direction-finder on the airplane is that the effect of side drift may be immediately noticed and corrected for by heading the airplane at the correct angle into the wind so that the airplane remains in the equi-signal zone, thus enabling the pilot to maintain a straight course along the line of direction to the transmitting coil beacon.—"Journal of the Franklin Institute."

10,000-R.P.M. ELECTRIC MOTOR.

An interesting and quite unusual type of electric motor is that described below. The machine, which is of 25 horsepower, is built to run at 10,000 R. P. M., being direct coupled to a plant requiring this speed. It is arranged with a vertical spindle. It operates on a direct-current 300-volt circuit, this non-standard voltage being that of the factory in which it is installed. The motor was built by Messrs. Mawdsley's, Limited, of Dursley, Gloucestershire, who have made a specialty of extra high-speed electric motors. The machine has been in use for some time and has performed in a very satisfactory way.

The running of an electric motor at 10,000 R.P.M. introduces many special difficulties which have to be allowed for in the design. Apart from the question of centrifugal forces, difficulties arise in connection with temperature rise owing to the core losses at the high-frequency, while the design of the bearing requires special attention. In the motor with which we are concerned the centrifugal force per pound at the circumference of the armature is nearly four tons, and it will be realized that careful choice of materials is necessary as well as good design, and workmanship. A special form of winding is used, instead of the usual barrel winding, and the ends are brought out close to the shaft, so that the connections from the armature to the commutator are arranged, as far as possible, in a radial plane. The effect is that the centrifugal force results in a longitudinal pull on the wires, and any bending effect is avoided. The wires were soldered into the commutator tags with the hardest solder obtainable, the commutator tags were then turned and insulated with mica and a steel ring was then shrunk on. The commutator has altogether three steel shrink rings.

Temperature rise is kept within proper limits by means of a fan, machined from a solid steel disc, and fixed on the armature shaft. This fan delivers jets of air on to the armature and commutator, and has proved so satisfactory that the temperature rise after a seven-hours' full load run is within the Engineering Standards Association specification limits. The brush gear is of Mawdsley's patent type, in which pressure is applied to the carbon brushes by coil springs. The peripheral speed of the commutator is approximately 12,000 feet a minute, and close grained electro-graphitic brushes have been found to give the best results. The commutation is perfect, in spite of the fact that the machine is required to run at speeds varying from 5,000 to 10,000 R.P.M. by means of shunt regulation, so that at the higher speed

the field is comparatively weak. The motor thrust bearing takes not only the weight of the armature, but additional weight due to the machine being driven. The conditions are such that the ordinary type of ball thrust bearing proved unsatisfactory owing to the heat generated by the friction of the balls on the cage, and special arrangements had to be made.—"Engineering," Feb. 22, 1924.

LENZ HYDRAULIC TRANSMISSION SYSTEM.

A New Type of Combined Reduction and Reversing Gear for Application to Marine Work.

In view of the many attempts now being made to adapt small high-speed engines for marine propelling plant, in place of those of the slow speed, direct coupled, reversible type, much interest is attached to a hydraulic device recently developed for this purpose by the well-known Austrian firm, the Grazer Wagen und Maschinen Fabrik.

Up to the present time in this country the electrical transmission system has found the greatest favor, but the system here described has indubitable advantages in reliability, cheapness and simplicity.

Several systems of hydraulic transmission are already in use for various purposes, such as the Foettinger transformer for ship work and the Williams-Janney gear for steering gears and winches, but the Lenz transmitter, as the new device is called, differs from these previous designs in using a wing or vane pump and a similar type of motor.

A complete installation is at present being fitted in a side-wheel tug, the transmitter in this case being inverted so that the driven or motor shaft is over the pump and engine shaft. This entails alteration in a few minor details of design only, and the trial results may be awaited with the greatest interest.

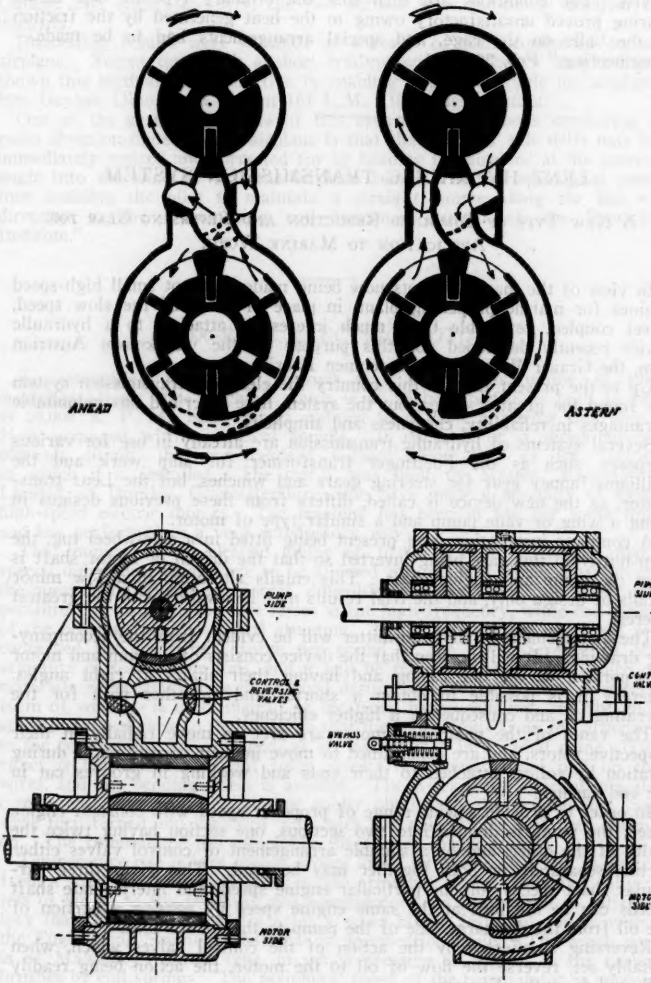
The arrangement of the transmitter will be evident from the accompanying drawings. It will be seen that the device consists of a pump and motor incorporated in the one casing and having their shafts at right angles, whereby it is possible to obtain a shorter and smoother path for the operating oil and consequently a higher efficiency.

The vanes of the pump and motor are free to move radially in their respective rotors, and are constrained to move in their correct paths during rotation by rollers attached to their ends and working in grooves cut in the end casings.

In order to provide a wide range of propeller speed with constant engine speed, the pump is divided into two sections, one section having twice the width of the other, and by a suitable arrangement of control valves either section separately or both together may be used. This gives three particular shaft speeds for one particular engine speed, and intermediate shaft speeds can be obtained at the same engine speed by passing a portion of the oil from the discharge side of the pump to the suction side.

Reversing is effected by the action of the control valves which, when suitably set, reverse the flow of oil to the motor, the action being readily followed from the diagram.

Relief valves are fitted wherever an accumulation of pressure is likely to occur, and arrangements are made for passing some of the oil through a cooler, as a certain amount of heat is developed by friction and churning.


AHEAD
ASTERN
PUMP SIDE
CONTROL & REVERSING VALVES
MOTOR SIDE
PUMP SIDE
CONTROL VALVE
BYEPASS VALVE
MOTOR SIDE

The compactness and simplicity of this transmitter when compared with a similar electrical system is readily apparent, and trials of an experimental unit have shown very favorable efficiencies.—"Motorship," Jan., 1924.

A NEW VARIABLE SPEED GEAR.

A new form of speed change gear embodying interesting constructive features, and designed for quick reversal with an infinite range of speed gradation from rest up to full speed in either direction, forms the subject of a recent article by Oscar Keller in our Swiss contemporary, the "Schweizerische Bauzeitung." The apparatus is of the fluid transmission type, and was invented by Heinrich Schneider, of Winterthur. The gear is protected by patents, and takes it name after its inventor. The first Schneider gear to be constructed was built at the Swiss Locomotive Factory, Winterthur, and after severe tests, it was applied to a large vertical boiler-plate bending machine with, we learn, very satisfactory results. The principle on which the gear works is illustrated diagrammatically by the drawings we reproduce in Figs. 5 and 6. It will be seen that the principal parts of the apparatus comprise two rotors placed one within the other, and each furnished with sliding vanes. The vanes of the outer rotor are in contact with an external housing fixed eccentrically with regard to the rotor, while the internal rotor is so suspended that its eccentricity with regard to the outer rotor may be varied between the two extreme positions—shown in Figs. 5 and 6 respectively. Assuming the relative position of the two rotors be that illustrated in Fig. 5, with the inner rotor turning in a clockwise direction, the fluid between the vanes will be compressed in the region of the working chambers A, and some of the fluid will pass through the slots in the body of the outer rotor and set up pressure in the working chambers B. At the same time suction will take place in the region of the working chambers C, and fluid will be drawn through the slots in the body of the outer rotor, setting up negative pressure in the region of the working chambers D. With these pressure conditions obtaining, the outer rotor moves in a clockwise direction or the same direction as that in which the inner rotor is turned. If now the internal rotor be moved up till it is concentric with the outer rotor there will be equal pressure on all the chambers and no fluid will pass from one rotor to the other, hence there will be no movement of the outer rotor relative to the inner. Again, if the internal rotor be now raised still further till it occupies the position shown in Fig. 6, a reversal of pressure conditions takes place, the chambers corresponding to D becoming a region of positive pressure in the outer rotor, and the chambers B a region of negative pressure. The fluid will now circulate the opposite way, the outer rotor moving in a counter clockwise direction, with the inner rotor turning clockwise as before. It will thus be seen that by altering the relative position of the rotors complete reversal and infinite gradation of speed from rest to full load is obtained in either direction. In Figs. 1 and 2 we show vertical and horizontal sections of the gear as actually constructed, while Figs. 3 and 4 indicate by means of cross-sections the relation of the rotors and the cast iron housing containing them and forming a base for the gear. We need hardly refer to the principle of the sliding vanes, since similar vanes have been in use for some years on compressor work with considerable success. The new feature would seem to be the combination of the two rotors and the means of altering their relative position one to the other, and of permitting a flow of fluid through the rotors and round equalizing passages in the casing so as to form a closed circuit.

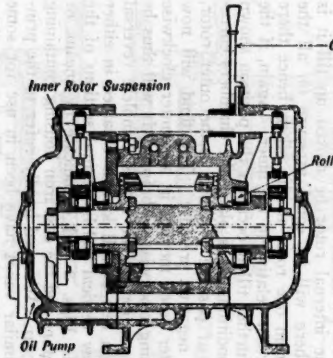


Fig.1.

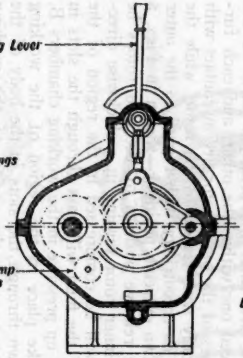


Fig.3.

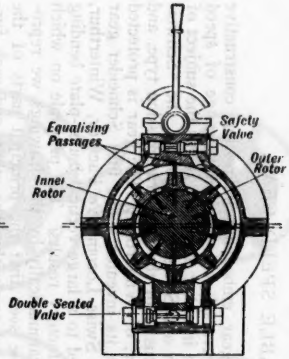


Fig.4.

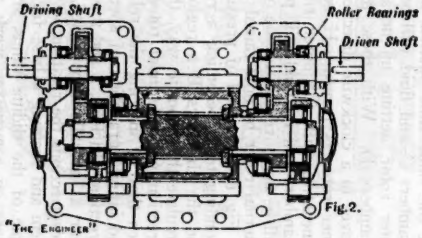


Fig.2.

"THE ENGINEER"

Fig.5.

Fig.6.

THE SCHNEIDER VARIABLE SPEED GEAR

The motion is transmitted from the driving shaft to the inner rotor, and from the outer rotor to the driven shaft by means of cut gears, and roller-type bearings are used throughout. Speed regulation is controlled by means of an operating lever pinned to a lay shaft having eccentric ends, to which the inner rotor housings are attached by means of suspension links. The operating lever moves over a sector showing the speed range in either direction. Fig. 4 serves to indicate the arrangement of equalizing passages and the safety valve. The gear-driven oil pump—shown in Fig. 1—draws its supply from the gear casing and delivers oil under pressure through a channel in the lower part of the gear, which terminates in a passage closed by a double-seated valve. The function of this valve is automatically to make the connection between the oil delivery passage and that side of the gear which at the moment happens to be the suction side. Among the advantages claimed for the gear we have described are its small and compact size, simplicity and noiseless working. It requires little attention, and would seem to fulfil all the requirements of a speed change gear for machine tools or lifting appliances. No particulars are given as to the efficiency of the gear or the ranges of speed for which such gears may be designed, but we understand that it may be built in the smallest sizes and for any desired range of speed reduction.—"The Engineer," March 7, 1924.

ASSOCIATION NOTES.

At the time of going to press the Annual dinner is about to take place. A report of the dinner will be made in the next issue of the JOURNAL.

The Council is anxious to increase the usefulness of the Society and to cover by the JOURNAL the entire field of Naval Engineering. The submission of articles for publication on any subject related to Engineering in the Marine field is requested, including aircraft.

The Society is composed of three classes of Members, which may not be generally known. Commissioned officers and ex-commissioned officers of the regular Navy, Marine Corps, and Coast Guard are eligible as Naval Members.

Civilians, who are especially interested in Naval matters or the Merchant Marine are eligible either as Associate or Civil Members.

The Council is undertaking a campaign for additional members and the Secretary-Treasurer will be glad to furnish further information to those interested.

The following members have joined the Society since the publication of the last JOURNAL:

NAVAL.

Antrobus, Charles, Lieutenant, U. S. N.
Coulton, Howard, Ensign, U. S. N.
Dierdorff, Ross A., Lieutenant, U. S. N.
Ellsberg, Edward, Lieutenant (CC), U. S. N.
Griffith, Cuthbert, Lieutenant, U. S. N.
Porter, Robert L., Lieutenant, U. S. N.
Strother, William J., Lieutenant, U. S. N.

Wakefield, William, Lieutenant, U. S. N.

Webb, John C., Lieutenant, U. S. N.

Webster, Frank F., Lieutenant, U. S. N.

Wiestling, Richard C., 49 Garden Place, Brooklyn Heights, N. Y.

Wilkinson, Ford L., Jr., Lieutenant, U. S. N.

ASSOCIATE.

Dahlstrand, J. Y., 37 Stevens St., Wellsville, N. Y.

Greene, Francis V., Lieutenant, U. S. N. R. F.

Nye, Robert G., 86 West Oakwood Place, Buffalo, N. Y.

Schaefer, Michael D., 601 Transportation Bldg., Washington, D. C.

Walter M McFarland.

WALTER MARTIN McFARLAND.

Our members and guests who attended the Annual dinner of the Society, held at the New Willard Hotel, Washington, D. C., May 17, 1924, had the pleasure of listening to an address on "The Old Navy and the New" by Mr. McFarland. It is regretted that space does not permit of including the address in this number of the JOURNAL. Mr. McFarland was for many years an officer in the Navy, has always been closely associated with the Navy, and is particularly qualified to speak on the subject of his address.

This distinguished engineer was born in Washington, D. C., August 5, 1859. He entered the U. S. Naval Academy as a cadet engineer and graduated with distinction in 1879. For a number of years thereafter he served as an officer in the Navy in various capacities both ashore and afloat. He was the right-hand man of the late Admiral Melville during a part of his long service as Engineer-in-Chief of the Navy. During his Naval career, he assisted in founding the American Society of Naval Engineers and is a loyal member and worker today. In 1899, he resigned his commission in the Navy to accept a position with the Westinghouse Electric and Manufacturing Company, where he was intimately associated with the late Mr. George Westinghouse. About fifteen years ago, he severed his connection with the Westinghouse Company and became Manager of the Marine Department of the Babcock and Wilcox Co., in which position he is now serving.

Mr. McFarland is President of the American Society of Naval Architects and Marine Engineers and has served on many important Engineering Committees.

The Navy takes pride in the knowledge that this nobleman was formerly an officer in the Navy and in his distinguished career both in and out of the Service. The American Society of Naval Engineers, whose good friend he is, wishes him many more years of happiness, service and prosperity.